DESIGN AND PERFORMANCE OF A VAPORIZER TYPE FUEL INLET SYSTEM

A Thesis

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SUMMARY

A vaporizing tube fuel inlet system was developed experimentally for a constant pressure 2 x 5 x 20 inch rectangular combustion chamber. Naphtha was used as the fuel at a design flow rate of .3#/min. The chamber was operated at a primary air/fuel ratio of 36/1, an over-all air/fuel ratio of 125/1, and a full section velocity of 125 ft/sec.

The resulting vaporizer required a substantially shorter primary combustion zone than a comparable spray injection system. The curves of efficiency versus fuel/air ratio were relatively flat showing little drop in efficiency as fuel flow rate was increased or decreased from peak conditions. The flame was blue having little luminosity. However, there was evidence that the good results were in part due to the high volatility of the fuel, and that insufficient heat was being transferred by the vaporizer.

The three quench air patterns tested were not effective in quenching combustion, and the resulting exhaust temperature profiles had high temperature peaks and were unsatisfactory for use with a turbine.

INTRODUCTION

Statement of the problem:

The design of combustion chambers for aviation gas turbine engines is slowly progressing through an evolutionary process which began on a trial and error basis,[1] and which has now reached a stage where much empirical data is available. However, the combustion process is so complex that it is unlikely that a purely analytical approach will be feasible in the near future. Consequently, there is a continuing and urgent need for test data which can provide a basis for further design improvements and can assist the theorist in achieving his goal.

The demands of aircraft designers for efficiency, high thrust, and low frontal area which have lead gas turbine engine designers to axial-flow compressors, straight-through flow paths, afterburners, and variable-area tail nozzles have also produced a very long engine. This length has reached such proportions that serious stability, maneuvering, and aircraft weight problems have been encountered, particularly in connection with fighter type aircraft. Consequently, all components of the engines are being scrutinized in an attempt to reduce length without increasing diameter. The combustion chamber offers some interesting possibilities in this respect.

A typical combustion chamber in an engine produced in the United States has atomizing fuel nozzles which spray the fuel into

the airstream. As the tiny fuel droplets are carried downstream by the primary, or combustion portion of the airstream, heat energy is transferred from high temperature surfaces and hot combustion gases to the air and fuel droplets. This transfer is aided by devices which produce vigorous mechanical mixing resulting in uniform small-scale high order turbulence.[2] The fuel becomes vaporized and the fuel-air mixture is heated to ignition temperature. After an ignition delay period, combustion begins.

A complex reaction and propagation process transports the flame front through the relatively low-speed primary air zone, and two other air flows come into the picture. There is the so-called secondary or quenching air which cools the very high temperature combustion products and provides a mixing action so that the gases leaving the combustion chambers are at a uniform temperature which is below the temperature which would damage the turbine blades or the turbine inlet guide vanes.

The third portion of the air flow is a protective blanket of air which cools the combustion chamber liner. This cooling air may be introduced in the forward or primary combustion zone or in the downstream zone where the secondary air is being introduced. Normally the cooling air is a very small percentage of the total flow, and after it performs its mission of protecting the critical surfaces it mixes with and becomes part of the primary or secondary flow. However, the degree of success with which the cooling air flow is handled may well determine the durability and success of

the entire combustion chamber.

The problem of reducing the length of the combustion chamber without increasing the diameter or sacrificing combustion efficiency can be attacked by several methods.[3] The one offering the best possibility is to eliminate that portion of the combustion chamber where the fuel is introduced and heated to ignition temperature thereby eliminating perhaps the greatest time lag in the ignition sequence, that of vaporizing the fuel.[9] A device by which this can be done is the vaporizing tube. This device is a cane or 'T' shaped tube leading into the front end of the combustion chamber, Fig. 1. The fuel and some of the primary air enter the combustion chamber through the tube which in turn has its outside surfaces exposed to the hot combustion gases within the chamber. As the fuel and air travel through the tube, the fuel is vaporized, mixed with air, preheated and discharged in an upstream direction. Thus it is possible to deliver fuel and air to the very front of the combustion chamber prepared for fast[7,8,9] and efficient combustion.

The vaporizing tube is not new. It was used by Frank Whittle in the first successful turbo-jet engine.[1] It is currently being used in several British turbo-jet and turbo-prop engines; the Armstrong Siddely Adder,[4] the Mamba,[5] and the Double Mamba.[6] The first applications were not too successful because any condition other than design condition resulted in rapid tube deterioration caused by carbon deposits from cracked fuel, or, in the other extreme,[1] delivery of unvaporized fuel into the combustion

chamber.

No production turbo-jet or turbo-prop engine in the United States has utilized vaporizing tubes. However, interest has recently developed[10] in response to the urgent need for reduction in length, and much design information is needed.

It was the purpose of this investigation to develop experimentally a vaporizer tube for use with a rectangular combustion chamber built by Janssen.[11] Several vaporizer tubes were tested at a predetermined design operating condition with one fuel (naphtha) to determine the combustion efficiency and minimum combustor length. Tests were also made to determine the variation of efficiency with air/fuel ratio.

Method of Approach:

The design operating condition was chosen to be similar to that of the naphtha tests made by Ryberg.[12] Thus it was planned to have test data from a system with a vaporizing type fuel inlet system which could be compared directly with the results from the conventional fuel spray system used by Ryberg.[12] The specifications of the naphtha used in the tests are reported in Appendix B. The design fuel flowrate for naphtha was .3#/min, primary air/fuel ratio 36/1, overall air/fuel ratio 125/1, and full section velocity 125 ft/sec. The basic airflow pattern including primary and secondary air is shown on Fig. 16.

A good vaporizer tube design was intended to provide not

only peak or optimum performance at the design operating condition as measured by the combustion efficiency and length of the primary combustion zone required, but also the curve of efficiency with air/fuel ratio should not show rapid loss of efficiency with variation from design condition.

Certain items in the design of the vaporizer tube including the tube diameter, wall thickness, wall material and tube shape were decided upon on the basis of preliminary considerations including heat transfer calculations, Appendix A. However, the actual heat transfer area required was determined on the basis of experimental tests. Three dimensions, shown on Fig. 1, are important in these tests. 'X' is the distance from the forward end of the combustion chamber to the vaporizer tube outlet. 'Y' is the distance from the tube outlet to the downstream end of the vaporizer tube. 'X' and 'Y' together are indicative of the total length of tube exposed to the flame temperatures within the combustor. 'Z' is the distance from the front of the combustion chamber to the most upstream quench air inlet. The 'Z' distance is the length of the primary combustion zone and when added to the length of the quench air zone, one obtains the total combustor length.

The optimum 'X' distance was determined experimentally using butane as the fuel. Under optimum delivery conditions the regular fuel, naphtha, would be delivered into the combustion chamber in a completely vaporized and superheated condition. Therefore, design conditions were simulated using a gaseous fuel

(butane) at a flow rate to give the same BTU/# of air as naphtha at its design flow rate. A series of tests was made to determine the variation of combustion efficiency with cane outlet position (X). 'Y' was held constant with the quench air at the most downstream position.

The effect of cane size (Y distance) on combustion efficiency was determined using naphtha as the test fuel at flow rate of .3 pound/minute. Five cane sizes were tested with Y ranging from 1 to 5 inches. Since it was desired to determine the length of combustion chamber required with a vaporizer system as compared with a fuel spray system, each cane was tested with the quench air in various positions ranging from all the way downstream to as far upstream as possible without interfering with the primary air pattern.

The effect of air/fuel ratio on combustion efficiency was also determined for each cane. The air flow rates and patterns were held constant at design conditions with the quench air in the downstream position. Then the fuel flow rate was varied from lean (unsteady operation) to rich (flame downstream into rake).

Three quench air patterns were tested, Figs. 16 and 17, in an effort to improve the effectiveness of the quench air without physically modifying the equipment. One of the patterns was similar to Ryberg,[12] Fig. 17C. One pattern attempted to produce better penetration by alternating top and bottom flow, Fig. 17B. The third attempted to produce better penetration by considerably increasing the flow rate, Fig. 17A.

TEST EQUIPMENT

General Description:

The combustion chamber used in this investigation was designed by Janssen,[11] and was intended to provide a predominantly two-dimensional process. It is a rectangular chamber five inches high, two inches wide and 20 inches long, Figs. 3 and 4. There are 48 rectangular air inlet ducts extending the full width of the chamber and located in symmetrical positions above and below the chamber center line. Each of these ducts contains a butterfly type control valve, a rectangular square edge orifice, a flow straightener, and an injection orifice. Thus the air supplied through the ducts may be measured and controlled so that the flow rates and flow patterns of the primary, and secondary air may be varied over a wide range.

The air for the primary and secondary flow was supplied by a centrifugal compressor having a capacity of twenty pounds of air per second and driven by a 165 horsepower Lycoming engine, Fig. 2. The engine was not equipped with any type of governor, but was controlled by a throttle in the control room, Fig. 11, through a hydraulic linkage. The air flow path was from the compressor through a circular duct into a 'Y' type rectangular manifold, and from the manifold into the 48 ducts leading into the combustion chamber. The vaporizer tube or cane under test, Fig. 5, was installed in the front end of the combustion chamber, Fig. 6.

The five test canes were weldments fabricated from size 5/8" O.D. X .028" wall type 347 stainless steel seamless tubing.

Naphtha was pumped to the cane by an electric motor driven Vickers constant displacement pump. Backpressure in the fuel line was controlled by a needle valve next to the point of discharge into the cane, Fig. 2. Discharge into the cane was through a $\frac{1}{4}$" flush opening in the side of the cane. The fuel flow rate was controlled by a second needle valve in the pump bypass system.

Air for the cane was provided by the building main air compressor system. This system has a storage capacity of 30 cubic feet at a pressure of 100 psig. The rate of flow of the cane air was controlled by a valve in the line.

The ignition technique developed by Ryberg[12] was used for all tests. Butane from a high pressure tank was allowed to flow through a pressure reducing regulator and a control valve to a nozzle adjacent to the cane in the front end of the compressor, Fig. 6. This location gave an easily ignitable mixture at the igniter plug and eliminated all false starts. The igniter plug was energised by a coil in the control room.

During tests in which butane was used as the test fuel, the liquid fuel line to the cane was replaced by the butane line which had been detached from the ignition nozzle. No special ignition technique was required.

Downstream of the combustor was a simple exhaust manifold system which led the combustion gases to an exhaust area. The

test room had a ceiling exhaust fan which caused an appreciable fresh air inflow through the test-room doors during tests and prevented any dangerous air contamination.

Instrumentation:

The primary and secondary air from the centrifugal compressor traveled through a 6 inch duct containing a 4 inch diameter circular square edged orifice (orifice #2, Fig. 2) with flange taps. The flange taps were located in accordance with ASME standards.[13] The pressure drop across the orifice was measured with a water manometer. The upstream gauge pressure was measured with a water manometer up to 46 inches. Higher gauge pressures were measured with a mercury manometer. The temperature was measured by an iron-constantine thermocouple. The flow rate was determined from Fig. 7.

The cane air passed through a 2 inch duct containing a 1.010 inch diameter circular square edge orifice (#3, Fig. 2) and flange taps.[13] Upstream and differential pressures were measured in the same manner as for orifice number 2. An iron-constantine thermocouple was used for the temperature. The flow rate was determined from Fig. 8.

The 48 combustion chamber inlet ducts had rectangular orifices for flow measurement which were calibrated by Janssen.[11] Janssen's manometer system involved measurement of pressure drop by water columns and upstream gauge pressure by a combination

of two water columns and one mercury column. This system was replaced by a simple water manometer system, Fig. 10, for two reasons. First, it was complicated to determine the upstream pressure using the old manometer system, because three fluctuating columns were involved. Second, a fluctuation or change in pressure on one column caused a measurable change in all of the other columns in the same manometer bank. These fluctuations occurred because the system was semi-closed and the reservoir surface area was not large enough with respect to the surface area of each individual manometer tube. Thus the old system made it difficult to set-up or change the primary or secondary air patterns. The new manometer system consists of a series of 52 independent 22 inch 'U' tubes for the measurement of pressure drop across the orifices and 34 tubes 46 inches long arranged in two manifolds (each with a large surface area reservoir) and used to measure upstream pressure. The main disadvantage of the new manometer system is its bulk and the large quantity of glass and rubber tubing required.

The boundaries of the combustion zone were estimated, using the same temperature basis as Ryberg,[12] from readings made with 39 chromel-alumel unshielded thermocouples Figs. 1, and 6. The values read on the potentiometer (iron-constantine compensated) were converted[14] to degrees Rankine by using Fig. 12.

The naphtha flow rate was measured by means of a roto-meter calibrated by the manufacturer to read gallons of gasoline per hour. It was calibrated for naphtha by using the weighing-tank

technique. A thirty minute calibration run was made at the design point to help insure accuracy. The results are plotted in Fig. 9.

The butane flow rate was determined by a Fisher and Porter flowmeter. This measuring device was not calibrated, but the flow rate was determined using the manufacturer's calibration.[15]

The average temperature rise achieved by the combustor was calculated on the basis of density, velocity and temperature readings obtained by utilizing a special rake installed in the diffuser section at a point .75 inches downstream of the chamber proper, Figs. 6, 13, and 14. This rake consisted of 7 pitot tubes made of .128" O.D. stainless steel tubing with a .06" inside diameter, and seven 20-gauge chromel-alumel shielded thermocouples. Figure 13 shows the fabrication method used. The individual pitot tubes were used as holders for the thermocouples. The tip portion of the pitot tube was machined down so that a section of oval ceramic tubing .48" long having two .10" diameter holes through it could be slipped on over the pitot tube tip and down as far as the machined shoulder, Fig. 15. A loose fit was required between the steel and the ceramic so that the ceramic would not be shattered when the pitot tube expanded because of exposure to the hot combustion gases. The other hole in the ceramic insulator served as a shield for the butt-welded thermocouple. After the thermocouple junction was located in the center of the insulator hole, the two ends of the wire were led out through notches in the front and back edges of the hole in the ceramic in such a manner that the wire was diagonally positioned

through the hole. This left sufficient room for an adequate gas flow through the opening. The two lead out wires were then strung through small two-hole ceramic insulators. The individual pitot tube with its adjacent thermocouple was then imbedded in wet shredded asbestos in the bottom half of an airfoil shaped fairing made of stainless steel. In order to permit disassembly for repair, aluminum foil was placed over the bottom half before more wet asbestos was piled on. The top half of the airfoil section, with appropriate leading edge slots to allow the thermocouples and pitot tubes to protrude, was next clamped in place as tightly as possible to squeeze out excess moisture. Care had to be taken to insure that none of the insulators were cracked during this tightening process. At this point a drilled jig was set over the tips of the pitot tubes to insure proper positioning, and the whole unit was baked in an oven until the asbestos was hard and dry. The unit was then installed so that the seven pitot tube-thermocouple pairs were each at the approximate center of a rectangle two inches wide and 1/7th of the height of the diffuser at the plane of measurement. The difference in pressure between the pitot tube total pressure and the static pressure[12] in the plane of measurement was measured with a water manometer. The thermocouple readings were made on a potentiometer compensated for iron-constantine and located out in the control room. Fig. 12 was used to convert the readings to degrees Rankine. The rake made it possible to read or check any position quickly, and no one was required to stand next to the

combustion chamber and adjust thermocouple or rake positions during a run. It is estimated that using the rake in place of the hand positioned thermocouple and pitot tube reduced the average time per run to one-half of its former value.

EXPERIMENTAL PROCEDURE

The experimental program consisted of a series of 102 individual tests. Each test was set up with some specified x, y, and z distance, an air flow rate, a primary and secondary air pattern, and a butane or naphtha flow rate. The measure of success of each test was the combustion efficiency, η . The thermal efficiency was defined for the purpose of these tests as the actual average temperature rise produced by the burner at the measuring station divided by the theoretical temperature rise. In addition to determining the thermal efficiency, it was also necessary to try to determine why the efficiency varied as it did. Therefore, other information was obtained from some of the tests including; area of combustion, visual description of the flame, pressure loss, and side plate condition.

The operational procedure for each test period began with a series of preliminaries including: turning on the air compressor and cutting in the high pressure line, bringing up the butane tank from the safe-storage area and connecting it up with its regulator, taking the barometric pressure, measuring the fuel temperature,

opening the valves from the storage tanks for the compressor motor fuel and for the test naphtha, connecting up and balancing the potentiometers, and starting the compressor engine.

Prior to each day's operation it was necessary to bleed the fuel line by opening the backpressure valve and letting fuel flow through the system at a high rate. Then the valve was closed until the backpressure was about 10 psig at design fuel flow rate.

After the compressor engine had run for a sufficient time so that it was well warmed up and could be depended upon to run fairly steadily, an approximate setting of the primary and secondary air patterns was made.

To insure a steady flow of cane air, it was turned on and allowed to flow until the compressed air tank pressure dropped and the compressor started. This procedure eliminated any compressed air which had had time to cool.

Ignition was accomplished using butane and following Ryberg's[12] technique. Then the liquid fuel was turned on and slowly brought up to operating flow rate while the butane was being tapered off. If it was intended to operate at either a very low or very high fuel flow rate, smooth combustion was first achieved using an intermediate naphtha flow rate. Then the fuel flow rate was either increased or decreased to the specified flow rate. The ignitor was turned off as soon as stable combustion had been achieved with the test fuel alone.

The primary and secondary air patterns were adjusted next

to compensate for the pressure differences existing between the cold and hot conditions. By this time the combustor had been operating long enough to approximate a steady state condition, and the manometers could be read to determine the conditions at orifices #2 and #3 and the seven rake positions. Temperature readings were made on the incoming air, the seven rake positions, and, in some cases, the 39 thermocouple positions within the chamber.

Without stopping combustion, either the fuel flow rate was altered or the air pattern was shifted or both, and another set of data taken. By using this technique it was possible to make as many as 13 tests in one afternoon. The only reasons for shutting down were to take the combustor side plate off and replace the cane, repair instrumentation, or refuel.

As experience was gained in operating the equipment, both speed and accuracy were improved. It was found desirable to have each person keep the same job throughout the tests. Utilization of the rake made it possible to have one operator concentrate on maintaining accurate control of fuel flow rate and engine RPM in addition to helping read and record temperatures. A second person operated the potentiometer selector switches and read temperatures. A third operator made all air flow adjustments and manometer readings.

In order to achieve a reasonable degree of success in reproducing test points, it was found to be necessary to maintain every controllable condition the same. This included such unlikely items as controlling the small difference between atmospheric and

test cell pressure and controlling fluctuations in this pressure difference by always leaving the same doors open and always having the exhaust fan on. It was found that accuracy could be improved by taking double or repeat readings on critical items such as rake temperature. If the second reading was sufficiently close to the first one, the average was used. If it was not close, additional readings were made.

The data were reduced by following the procedure of Appendix C, sample calculations. The results of each test were expressed in terms of the combustion efficiency. Sometimes the flame pattern was plotted. Accuracy of the tests is estimated in Appendix D, estimated errors.

RESULTS AND DISCUSSION

Controlling the Air Flow Rates:

Review of the calibration procedure[11] used on the 48 air inlet orifices revealed: 1. Only 19 of the orifices were calibrated, 2. There was an appreciable variation among those tested, and the resultant calibration curve was estimated to be accurate to only ± 5% at the time the tests were made, and 3. Calibration included compensation for whatever loss there was through the duct slip-joints at the time of calibration.

In view of these facts and considering that two test projects had been completed after calibration, it was concluded that the original calibration data could not be relied upon to give the true mass flow even within ± 5%. The time schedule of the test equipment was such as to preclude recalibration. However, it was concluded that the purpose of the test project would be served if the calibration were approximate provided only that mass flows could be reproduced on a given inlet duct by maintaining a given pressure drop and density. Consequently, the primary and secondary air patterns are specified in terms of inches of water drop rather than in terms of mass flow (changes in density were negligible), Figs. 16 and 17. The mass flow rates resulting from these settings served to establish the flow patterns and the division between primary and secondary air. The overall air/fuel ratio was controlled by means of the #2 and #3 circular orifices and the fuel flowrator.

Quench Air Patterns:

An effective quench air pattern is expected to reduce the combustion temperatures to a uniform value with a minimum pressure loss and in a minimum of combustor length. An inspection of flame patterns from tests by Ryberg[12] revealed that his pattern, similar to Figs. 16 and 17C, was not effective in quenching combustion. The incoming air streams, opposed top and bottom, did not seem to have any ability to penetrate. The active combustion area was somewhat compressed or stratified into a middle area, but combustion continued downstream of the quench air. Under these conditions, the efficiency and the quench air position are not a valid measure of the length of the primary or secondary zones required, and the exit area temperature profile has a pronounced high temperature peak. This can be seen in Fig. 20. As the quench air pattern was moved forward from the downstream position (14 1/8") to the upstream position (6 5/8") the area of combustion between the quench air and the rake was appreciable in area and variable in size. The only method which would have given a realistic indication of the variation of efficiency with combustor length would have been one in which the temperature rise was based upon measurements made just downstream of the last quench air inlet. This technique would have required measuring equipment in the combustion chamber and could not be followed.

Two other quench air patterns were tested in an effort to improve both the quenching effect and the temperature reduction

effect. The first of these patterns, Fig. 17B, attempted to gain better penetration by not having the incoming air streams directly opposed. Two ducts on the bottom were set for a 5" drop opposite two .1" drop (bleed air) ducts on top. This was followed by two 5" ducts on top etc. This pattern increased the length of the quench air portion from 3 3/4" for the standard pattern to 6 3/4". However, it was hoped that effective penetration would be accomplished. Then the data could be used to establish the necessary length of the primary zone. However, the alternating pattern showed little, if any, improvement over the standard pattern.

The remaining pattern, Fig. 17A was similar to the standard pattern except that the flow rate was increased so that the drop across the orifice was increased to 11 inches and the number of orifices was reduced to keep the same overall air flow. This increased the injection velocity about 35%. Eleven inches was selected since it was about the maximum drop which could be maintained under steady conditions. The eleven inch pattern did not show any improvement over the standard pattern as regards effectiveness in quenching the combustion.

It should be noted that variation of the manner of introducing the quench air was limited by the requirement that the existing chamber could not be modified. This limitation was assumed because another investigation had been scheduled for that purpose.[17]

Since neither of the alternative patterns were effective,

all of the cane design tests were conducted with the standard 6 inch pattern, Fig. 17C. The manner in which this pattern could be shifted is shown in Fig. 20.

The Butane Tests:

The optimum X distance was determined by a series of five tests with butane gas as the test fuel. A 'T' cane with upstream sections 2 inches long was used. Design fuel flow rate, and standard primary and secondary air patterns were used, Fig. 16, with the quench air all the way downstream. The X distance was varied in one inch increments from 7 to 3 inches. These tests are plotted in Fig. 18. The curve of efficiency versus cane outlet position showed practically no variation in efficiency as the cane outlet was moved forward from 7 to 4 inches, but the 3 inch setting showed an appreciable drop. Therefore, a cane outlet position 4 inches downstream was selected as the X distance to be held constant for all subsequent naphtha tests. This position showed a high efficiency combined with a small X distance.

Use of Exit Mass in Computing Efficiency:

The butane tests were made before the exit rake had been installed. Therefore, the temperature and velocity were measured by the hand positioned unshielded thermocouple and pitot tube used by Ryberg.[12] However, Ryberg's[12] ten vertical positions and horizontal traverse were not used. In anticipation of the seven position rake, readings were made at the same seven positions in

the vertical plane as would later be used with the rake.

It was noticed that the total mass flow rate indicated by the exhaust traverse ($\sum_1^7 \rho_i V_i A_i$) was less than the summation of the inflowing air mass and fuel mass rates. Computation of the combustion efficiency based on the two sets of mass data resulted in the two sets of points shown on Fig. 18. Since the incoming mass flow was higher, a lower ΔT_{th} and in turn a higher η resulted. The lower mass flow indicated at the exhaust gave lower and more reasonable efficiencies. The actual temperature rise had to be based upon $\rho_i V_i$ and it was concluded that a more accurate efficiency would be indicated if the same mass flow values were used to compute both the theoretical and actual temperature rises. Therefore, on all subsequent tests, the reported efficiency was based upon the exhaust mass.

The reason for the difference between the inflow and exhaust mass flow rates can be explained in part by the fact that after expansion of parts due to the combustion temperatures and after vibration during a run, there was known leakage along the slip joints of the 48 air inlet ducts, through various pressure, thermocouple, and observation openings, and along the gaskets. Another error was introduced by the fact that the exhaust survey was made at seven discreet points and applied over the entire area. Also, the temperatures and pressures indicated may be in error due to position, radiation, conduction, non-axial flow, etc. It should be noted that the change in efficiency due to using the exhaust

mass was much greater for butane tests without the rake (about 12%) than it was for the naphtha tests utilizing the shielded rake (about 2%), see Fig. 24. In fact this figure shows that in some cases the exhaust mass showed an apparent increase!

Elimination of a Tendency to Surge:

The first naphtha tests were made with a 5 inch cane (x = 4", y = 5"). This cane was fabricated with a turbulence producing 'screen' which consisted of an area of closely spaced holes in tube A, Fig. 6, made with a number 60 drill. The entire area of tube A covered by tube B when the two tubes were joined together by weld C was filled with these holes. The cane thus fabricated had been satisfactory for the butane tests. However, when naphtha was used, there was produced a very pronounced and uncontrollable surge. First, there was a small combustion area near the cane outlet which would rapidly build up and move downstream and out the tailpipe with a roar (one section of the tailpipe had been removed for visual observation purposes). Then it would subside rapidly to the original small area. The repeated build up of the flame was paralleled by a build up of fuel backpressure. Therefore, it was concluded that initially the tube B was not hot enough to vaporize the liquid fuel which tended to collect at the orifice section and choke the airflow. Then the backpressure would increase and drive a slug of fuel through the screen. The expanding flame area would heat the entire cane

vaporizing all of the remaining collected fuel. The resulting fuel surge would drive the flame downstream until the cane was no longer immersed. Thus the collected fuel supply would become exhausted, the backpressure would drop, the cane would cool off, and the liquid fuel build-up would begin again. The cane was remodeled by drilling out the turbulence 'screen'. This eliminated the tendency to surge. However, there remained a tendency for a small amount of fuel fluctuation. This was traced to a spring loaded bypass valve in the liquid fuel pump system. It was replaced by a simple needle valve. Then the fuel flow rate could be set and maintained with a degree of accuracy commensurate with the accuracy of the fuel flowrator.

The Naphtha Tests:

Two series of tests were conducted with a 5 inch cane ($x = 4"$, $y = 5"$). A series of 5 tests was made with design primary and secondary air patterns with the quench air at the downstream position, Fig. 22A. Fuel flow rate was varied from 2.0 to 3.75 gallons per hour. These curves were plotted with efficiency as the ordinate versus fuel/air ratio in terms of theoretical BTU's liberated per pound of air as the abscissa. This was done so that a better comparison could be made with tests on other fuels.[18]

A series of 17 tests was made with design fuel flow rate and standard primary and secondary air patterns at an overall air/fuel ratio of 131:1. The quench air position was varied from 14 to

6.5 inches, Fig. 22B. The efficiency showed a steady increase as the quench air was moved forward reaching a maximum of 104% with the most upstream quench air just downstream of the last primary air inlet. This test series was the first one made with a shifting quench air pattern and was disappointing in the amount of scatter involved. Tests which apparently were valid varied as much as 9% from the curve. However, during these tests it became evident that certain items, at first considered unimportant, required control. Furthermore, the operators acquired skill in making finer adjustments and in reading mean values. Consequently, subsequent tests showed less deviation.

The plan had been to try to make the first cane too long. Then it would be a simple matter, after each test, to cut off an inch from each upstream section for the next test. However, the 5" cane did not show any signs that it had been overheated during the tests. There was no sign of coking or other fuel deterioration. Therefore, at this point it looked as though a longer cane would be needed. However, the cane was cut to 4 inches, ($x = 4"$, $y = 4"$) and 6 tests were made for fuel/air ratio and 8 tests for length of primary combustion zone, Fig. 23 and 24. The curves were similar to those of the 5 inch cane except that the efficiencies were higher! This trend continued through the three, two, and one inch canes, Figs. 25, 26, 27, 28, and 19. No tests were made with Y less than one inch since it was reasoned that with a smaller distance the flow would lose direction.

Figures 29 and 30 are a summary of all of the naphtha tests. Figure 29 shows that as the length of the upstream sections of the cane was decreased, the efficiency increased. The rich limit ends of the curves show a fairly uniform trend toward increase in maximum fuel/air ratio with decrease in cane size. The lean-limit ends of the curves are more erratic. The characteristics of the combustion in this region seemed to be more dependent on the manner of starting. That is, it seemed that there could be more than one stable condition of operation. This was given evidence by the fact that in a couple instances there was a bright spot on the combustor side plate while on a repeat run there would be no spot at all. Thus indicating changes in flow pattern and flame location. All the curves showed a relatively small change in efficiency over a range of fuel flow rate from lean limit to about double the lean value.

The efficiency levels are all relative and have little meaning as far as absolute value is concerned, Appendix C. Obviously the 1" cane could not have exceeded 100% let alone 110%. However, the curves do have meaning with respect to each other, and the surprising thing is that they show that the heat transfer area of the cane is quite unimportant as far as combustion of naphtha is concerned. To prove this point, a run was made without any cane! A straight section of tubing was inserted in the front of the chamber in such a manner that only <u>one inch</u> of it protruded into the chamber. Combustion was started in the usual manner using butane and tapering off to naphtha alone. The one-inch stub produced a

blue flame, and an efficiency of 83.9% at design fuel flow rate with the quench air all the way downstream. This was better than the peak efficiency for the 5" cane for the downstream quench air position. This led to the conclusion that the heat transfer rate, even with the 5" cane, was much lower than expected or required. Thus the preliminary heat transfer calculations may have been an indication of the real problem involved.

The curves of efficiency versus length of primary combustion zone, Fig. 30, show an increase in efficiency with a decrease in cane length with the exception of the 2" cane. The 2" cane has lower efficiency than the 3" cane in the range from 8 to 13 inches. The curves also show an increase in efficiency with decrease in length of primary zone. None of the curves show a drop-off in this respect. As pointed out previously, the terminology, 'length of primary combustion zone', is somewhat misleading because combustion continues through and downstream of the quench air zone. However, these tests were conducted in a manner comparable with Ryberg's tests[12] and indicate that a shorter primary combustion zone can be used with a vaporizer tube than with a spray nozzle. His curve shows a decided break in efficiency at about 9.3 inches. The efficiencies are not directly comparable because a different measuring technique and shielded thermocouples were used on the present tests.

The 2" cane was selected as the design cane because it showed a superiority over the 3" cane on all fuel/air ratio tests

and gave stable operation over a much wider range. It also showed superiority over the other canes at both ends of the combustor length curves. The 1" cane was not selected because it was desired to have extra heat transfer surface for tests of less volatile fuels.[18] Also, there was some question as to the validity of the 1" cane efficiencies as compared with the other canes, because, the combustor was modified after the 2" cane tests and prior to the 1" cane tests and it was impossible to restore the combustor to its original condition.

Combustion Patterns and Exhaust Profiles:

Figure 20 shows typical combustion area patterns and exhaust temperature profiles for various quench air positions. As the quench air was moved forward with a constant fuel flow rate, the forward portion of the pattern remained essentially constant, the area of active combustion between the quench air and the rake became larger, and the efficiency increased. The increase in efficiency probably was due to the fact that the quench air gradually mixed in and tended to help complete the combustion rather than penetrating and causing an abrupt quench. The temperature profile had a lower peak and a somewhat better distribution at the forward positions because there was a greater distance in which turbulence could cause mixing upstream of the rake. However, none of the temperature profiles were uniform enough to be satisfactory for use with a turbine.

Figure 21 shows the effect of variation in fuel flow. Both

the total combustion area and the peak exhaust temperature increased with fuel flow rate. This figure shows a pronounced tendency for more combustion in the lower portion of the chamber. This probably indicates the effect of the action of gravity on the fuel droplets during passage through the center tube, and prior to vaporization. More than 50% of the fuel must be coming out of the bottom leg of the T. This is also indicated by the fact that the combustion in the area upstream of the bottom outlet acted as if it were near the rich limit, and the forward portion of the flame boundary moved downstream as the fuel flow rate was increased. Again the temperature profiles were too peaked to be satisfactory for turbine use.

CONCLUSIONS

1. The T cane type vaporizer tube required a shorter primary combustion zone than a spray nozzle system operated under comparable conditions.
2. The efficiency versus fuel/air ratio curves were relatively flat showing little loss in efficiency as fuel flow rate was increased or decreased from peak conditions.
3. The combustion was very clean. The flame was blue, had little luminosity, and there were no traces of coking or carbon build up at any time.
4. The required fuel pressure can be very low. Essentially the pressure required is that of the fuel pumping and regulating system.
5. Neither the standard quench air pattern nor the two alternate patterns produced satisfactory quenching.
6. The exhaust temperature profiles were unsuitable for use with a turbine.
7. The exhaust area survey which was made was not adequate enough to establish the actual combustion efficiency. However, there were indications that a high combustion efficiency was produced by the vaporizer tube.
8. The heat transfered by the canes tested was secondary in effect on combustion efficiency to other factors such as cane position and combustion chamber flow characteristics.

9. The value of a vaporizer tube system could be more effectively studied by using a fuel less volatile than naphtha, one which would require more effective vaporization for satisfactory combustion.

RECOMMENDATIONS

1. The present equipment should be replaced with a unit which is effectively sealed against leakage.
2. The system should include provisions for installing test combustion chambers and liners of various designs and lengths.
3. A temperature rake in the flow just downstream of the chamber could be used to indicate temperature profiles and to determine whether or not active combustion had ceased. Total temperature rise should be measured a distance downstream where more adequate mixing had occurred.
4. Investigations of vaporizer tubes should include measurement of tube wall temperatures so that the amount of heat being transferred can be ascertained more accurately.
5. Provision for altering the flow condition around the cane exterior should be considered so that heat transfer rates can be altered if they are found to be low.

NOMENCLATURE

A . . . . . Area, feet^2.

a . . . . . Acoustic Velocity, ft./sec.

$\overline{C_p}$ . . . . Mean specific heat at constant pressure, BTU/#^oF.

D . . . . . Diameter.

H . . . . . Enthalpy.

h_{12} . . . . Film coefficient of heat transfer between area 1 and surface 2, BTU/hr.ft.^{2} ^oF.

K . . . . . Thermal conductivity, BTU/hr.ft. ^oF.

L . . . . . Length.

M . . . . . Mach number, V/a.

$\dot{M}$ . . . . . Mass flow, #/sec.

N_{Pr} . . . . Prandtl number.

N_{Re} . . . . Reynold's number.

P . . . . . Pressure.

Q_L . . . . Lower heating value of a fuel.

Q_L' . . . . Lower heating value of a fuel minus latent heat of vaporization.

sg . . . . Specific gravity.

r . . . . . Radius.

T . . . . . Temperature, ^oR.

t . . . . . Temperature, ^oF.

U . . . . . Overall heat-transfer, coefficient, BTU/hr.ft.^{2} ^oF.

V . . . . . Velocity, ft/sec.

X . . . . . Distance from front of combustor to cane outlet, inches.

Y . . . . . Length of upstream sections of cane, inches.

Z . . . . . Distance from front of combustor to first quench air inlet port, inches.

η . . . . . Combustion efficiency.

ρ . . . . . Density, #m/ft^3.

Δ . . . . . Increment.

μ . . . . . Dynamic viscosity, #hr./ft^2.

Subscripts

a . . . . . Air.

act. . . . Actual.

Abs. . . . Absolute.

bp . . . . Boiling point.

f . . . . . Fuel.

i . . . . . Any point.

m . . . . . Mixture of air and fuel.

theo . . . Theoretical.

O.D. . . . Outside diameter.

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APPENDIX A

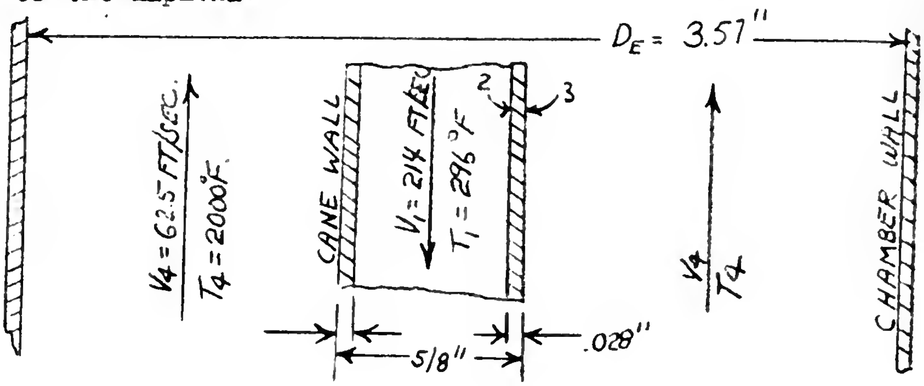
Preliminary Design of a Vaporizer Tube

The usual shape of a vaporizer tube is similar to an ordinary walking cane[4,5,6]. A round tube is fastened at the forward end of the combustion chamber. At the downstream end of the straight portion of the tube there is a 180^{o} bend leading to a short straight upstream section. This design is convenient for annular combustion chambers because the tubes can be arranged in such a manner that the outlet of one tube is adjacent to the straight down-portion of the next tube, and so on around the circumference of the chamber. Thus the inlet portion of a tube is always bathed in the very hot flame from the next tube's outlet. However, the combustion chamber utilized for these tests has a cross section of 2 x 5 inches. Thus only one cane could be used. Therefore, a "T" shape was selected, Figs. 5 and 6. This configuration was intended to give a more uniform distribution of combustible mixture over the long dimension of the chamber and thereby help provide the intended two-dimensional characteristics of the burner. Furthermore, it was hoped that the "T" configuration would provide a high average temperature over the entire external surface of the cane.

The cane material, type 347 stainless steel, was selected as an available material which would be tough and durable under the expected conditions of high temperature. The cane diameter, 5/8", was selected as a compromise between a large diameter for slower

flow with more time for heat transfer, and a smaller diameter to minimize the volume of the combustor lost due to tube immersion. The cane wall thickness was found to be relatively unimportant in the heat transfer process. Hence, a .028" wall thickness was chosen as a compromise between a thin wall for better heat transfer and a thick wall for ease of weldability.

Heat transfer calculations were based on the combined effect of conduction and forced convection. The net effect of radiation was not included because of the difficulty of assessing the effect of radiation toward the cane surface from gaseous combustion products (low luminosity blue flame) in competition with the surrounding rather dark-gray low temperature surfaces of the chamber. It was assumed that neglecting radiation effects was being very conservative (a larger required heat transfer area would be indicated). Another assumption on the conservative side was that the heat energy was ~~that the heat energy was~~ being transferred to dry air within the cane, while the quantity of heat energy required per unit time was that necessary to heat the air and naphtha to the ignition temperature of the naphtha.



The heat transfer calculations were made in three steps using the dimensions, velocities and temperatures shown in the preceding sketch. Firstly, the necessary rate of heat transfer was estimated. Secondly, the heat transfer coefficients from 1 to 2, 2 to 3, and 3 to 4 were estimated, and thirdly, the outside surface area of the cane was computed. This in turn was expressed as length of tubing since the tube diameter was known.

The heat required to raise 1.32 #air/min. and .3#naphtha/min. from 60°F to the ignition temperature of naphtha (531°F) was computed as follows:

$$(\Delta \dot{H})_{air} = \overline{C_p}\,(T_4 - T_1)\dot{M}_a = .2434(531-60)\ 1.32 = 151.3 \text{ BTU/min.}$$

$$(\Delta \dot{H})_{naphtha} = \left[\Delta h \text{ to boiling point} + \Delta h \text{ vaporization} + \Delta h \text{ to ignition point}\right]\dot{M}_f$$

$$(\Delta \dot{H})_{naphtha} = \left[.493(254-60) + (139) + .50(531-254)\right].3 = 112.0 \text{ BTU/min}$$

Rate of heat energy transfer required = (151.3 + 112.0)(60) = <u>15,798 BTU/hr.</u>

The inner film coefficient for heating of fluids in turbulent flow through pipes was computed using equation VIII-1,[16] as follows:

$$h_{12} = \frac{K}{D}(.023)\ (N_{Re})^{.8}(N_{Pr})^{.4} = \frac{K}{D}(.023)\left(\frac{VD\rho}{\mu}\right)^{.8}(N_{Pr})^{.4}$$

$$h_{12} = \frac{.0199}{.0474}\ (.023)(28,400)^{.8}(.703)^{.4} = 30.6 \text{ BTU/hr.ft}^2\ {}^{\circ}\text{F.}$$

The outer film coefficient for cooling of fluids in turbulent flow through pipes was computed by assuming that the fluid was flowing within a circular pipe with an area equal to that of the cross section of the chamber, and with a velocity equal to one-half of the full section velocity. One half of the temperature drop was assumed to

have occurred between the combustion gas and the outside surface of the cane. Equation VIII-2[16] was used:

$$h_{34} = \frac{K}{D}(.027)\left(\frac{VD\rho}{\mu}\right)^{0.8}\left(\frac{\mu C_p}{K}\right)^{.333}\left(\frac{U}{U_S}\right)^{.14}$$

$$h_{34} = .053(.027)\left[\frac{(225.000)(.297)(.000502)}{.00378}\right]^{.8}\left[\frac{(.00378)(9.17)}{.053}\right]^{.333}\left[\frac{.00378}{.00295}\right]^{.14}$$

$$= 6.25 \text{ BTU/hr.ft.}^2\ {}^{o}\text{F.}$$

The overall heat transfer coefficient based on the outer area of the cane was computed as follows using equation IX-40a:[16]

$$U_3 = \frac{1}{\dfrac{r_3}{h_{12}\, r_2} + \dfrac{r_3 \ln(r_3/r_2)}{K_{23}} + \dfrac{1}{h_{34}}}$$

$$U_3 = \frac{1}{\dfrac{.313}{30.6(.285)} + \dfrac{(.313/12)\ln(.313/.285)}{13} + \dfrac{1}{6.26}} = 5.11 \text{ BTU/hr.ft.}^2\ {}^{o}\text{F}$$

Once the overall heat transfer coefficient and the required heat transfer rate were known, it was possible to compute the length of tube required under the assumed conditions:

$$\dot{HM} = U_3\ 2\pi r_3\ L_3\ (T_1 - T_4)$$

$$L_3 = \frac{\sum \Delta \dot{HM}}{U_3\ 2\pi r_3\ (T_1 - T_4)} = \frac{15.798}{(5.11)2\pi(.313/12)(2000-60)} = 9.7 \text{ ft.}$$

This length is impossible to utilize and originally these results were thought to indicate that the assumptions required to make the above computations introduced errors of such a magnitude as to make the results of little value. However, the test results seem to indicate that actual heat transfer rates were much lower than expected.

Other designs[4,5,6,10] have been successful using much smaller canes. Therefore, in spite of the large surface area indicated by the simplified calculations, it was decided to procede on an experimental basis. A cane was selected having a total length within the chamber of nine inches (45% of total chamber length) with two upstream sections 5 inches in length (this design is called a 5" cane). It was intended to test the cane; then cut off a 1" section of each of the upstream arms to make a 4" cane. If the tests of the shortened cane indicated lower efficiency, a cane longer than the 5" cane would be tested. If it showed higher efficiency than the 5" cane, it would be shortened again, and so on.

APPENDIX B

Fuel Specifications

The fuel used for these tests was Varnish Maker's and Painter's grade Naphtha supplied by the Pure Oil Company. An A.S.T.M. distillation[18] was made on a sample drawn from the test storage barrel and the results are plotted in Fig. 31. Since this fuel was a pure cut of crude with no additives, the net heating value, latent heat of vaporization, specific heat, and weight percentages of hydrogen and carbon were estimated from equations using the specific gravity.[19]

$$\begin{aligned} \text{Net Heating Value} &= 19{,}960 - \left[3{,}780\,(sg)^2\right] + (1{,}362 \times sg) \\ &= 19{,}960 - \left[3{,}780\,(.7624)^2\right] + (1{,}362 \times .7624) \\ &= 18{,}800 \text{ BTU/\#.} \end{aligned}$$

$$\begin{aligned} \text{Latent Heat of Vaporization} &= \frac{110.9 - (.09 \times {}^{\circ}\text{F.})}{sg} \\ &= \frac{110.9 - (.09 \times 60)}{.7624} = 139.3 \text{ BTU/\#} \end{aligned}$$

$$\text{Wt.\% of } H_2 = 26 - (15 \times sg) = 26 - (15 \times .7624) = 14.56\%$$

$$\text{Wt.\% of C} = 100 - (\text{Wt.\%}H_2) = 100 - (14.56) = 85.44\%$$

$$C_p = \frac{.388 + (.00045 \times {}^{\circ}\text{F})}{\sqrt{sg}} = \frac{.388 + (.00045 \times 60)}{\sqrt{.7624}} = .476 \text{ BTU/\#}^{\circ}\text{F}$$

APPENDIX C

Sample Calculations

Basic Equations:

The assumptions of steady flow and no heat loss through the combustor walls reduce the energy equation to a simple form defining theoretical temperature rise: $Q_L' = \overline{C_p}(\Delta T_{theo.})$. This equation can, in turn, be used as the basic definition for the combustion efficiency: $\eta Q_L' = \overline{C_p}(\Delta T_{act.})$.

Sample Calculations:

1. Airflow calculation for #2 and #3 orifices,[13] Figs. 7 and 8.

$$\dot{M}_a = (.668)(A_2 KEY)\sqrt{\rho \Delta P}$$

$$T = 543^oR, \quad P = 30.94'' \text{ Hg}, \quad P = 4.60'' \text{ } H_2O$$

$$\rho = P/RT = .0756 \text{ \#/ft}^3$$

$$\dot{M}_a = .6370 \text{ \#/sec}$$

2. Velocity calculation

$$\Delta P = 10.02'' \text{ } H_2O, \ \rho = .0175 \text{ \#/ft}^3$$

$$V = \sqrt{\frac{\Delta P('' \, H_2O)}{\rho \, (\#/ft^3)} \, 334.9} = \sqrt{\frac{(10.02)(334.9)}{.0175}} = 437.7 \text{ ft/sec}$$

3. Average temperature rise

$$\text{Average exit temp.} = \frac{\sum_1^7 V_i T_i \rho_i}{\sum_1^7 V_i \rho_i} = \frac{71,325}{57.18} = 1248^oR$$

$$\text{Average Inlet Temp.} = \frac{M_f \, T_f \, (in) + \sum_2^3 M_a \, T_a \, (in)}{M_f + \sum_2^3 M_a} = 542^oR$$

$$\Delta T_{act} = T_{ave.}(out) - T_{ave.}(in) = 1248 - 542 = 706^oF$$

4. $\dot{M}_{(m)} \left(\overline{C_{p}}_{(m)} \right)$

$\overline{C_p}$ air = .2473, Fig. 32 for 542°R to 1248°R

$\overline{C_p}$ fuel = .498 BTU/#°F

$(\dot{M}_m)(\overline{C_{p_m}}) = (\dot{M}_f)(\overline{C_{pf}}) + (\dot{M}_a)(\overline{C_{p_a}}) = (.00592)(.498) + (.6315)(.2473)$

$(\dot{M}_m)(\overline{C_{p_m}}) = .1593$ BTU/°F Sec.

5. Q_L' fuel

$Q_L' = Q_L$ - Latent heat of vaporization = 18,769 BTU/#

6. $\Delta T_{theo.}$

$$\Delta T_{theo.} = \frac{\dot{M}_f \times Q_L'}{\dot{M}_m \times \overline{C_{p_m}}} = \frac{.00592(\#/sec)\ 18,769\ BTU/\#}{.1593(BTU/sec\ ^\circ F)} = 698^\circ F$$

7. Combustion efficiency

$$\eta = \frac{\Delta T_{act.}}{\Delta T_{theo.}} = \frac{706}{698} = 101.1\%$$

APPENDIX D

Estimated Accuracy

Reading Errors:

1. The average error in reading total pressure, upstream static pressure, and pressure differences was $\pm$ .05" H_2O.

2. Liquid fuel flow reading error was $\pm$ .025 gph.

3. Temperatures were read to $\pm$ 2°F.

Fluctuations:

1. Fluctuations of P and P on #2 and #3 orifices were not measurable.

2. Compressor RPM = $\pm$ 5.

3. Rake Δ P readings averaged $\pm$ .1" H_2O.

4. Rake temperature readings $\pm$ 4°F (actual fluctuations may have been larger. The indicated readings may have been lower due to thermocouple lag.)

Accuracy:

1. Airflow was measured with orifices fabricated in accordance with ASME power test codes[13] and are assumed accurate to $\pm$ 2%.

2. Fuel flow rate was controlled to $\pm$ 1%.

3. Temperatures measured by thermocouples within the chamber were quite inaccurate. Accuracy is estimated to be on the order of +0 to -10%.

4. Rake temperature measurements:

Accurate calibration of the thermocouples used in the rake was not accomplished at the temperatures to which they were exposed during the tests, because such calibration would have cracked or coated the ceramic insulation and ruined the rake. However, a test of each thermocouple in boiling water made after completion of all the tests yielded readings with a maximum variation of 1.5°F. from the average. Thus all thermocouples were shown to have survived the tests in good mechanical condition. The ceramic shields were designed to give the junction a very small view either upstream or downstream, Fig. 15. The total included angle, not allowing for blockage by the wire, being less than 24°. Furthermore, visual observation of the flame during tests showed it to be a low luminosity blue with very little variation from test to test. Visual observation of the thermocouple shields and the downstream area viewed by the junction showed that once they had acquired the grayish deposits from an initial test, they did not change in color or condition. A study of the exit profiles showed that temperature readings on comparative runs were quite close for individual thermocouples. Therefore, while the reported temperatures could be in error as high as 3 to 5% in absolute value, the change in error from test to test was estimated to amount to less than 1% of the reported temperature. In other words, the accuracy of the results should be very good for comparing one test with another to determine which was better.

5. The pitot tube velocities were estimated to be accurate to within $\pm$ 2% since the maximum test Mach numbers were very low (.4 max.).

6. All of the errors heretofore mentioned would cause average errors in efficiency in the order of $\pm$ 3% or less. Therefore, the values are considered to be good for comparison purposes. However, the largest single error involved was in assuming that flow was truly two-dimensional and basing the entire temperature rise on measurements taken on the center vertical section of flow. Horizontal profile measurements by Ryberg[12] showed that temperatures near the wall were considerably lower than at the midpoint. The steepness of the temperature and velocity gradients on both the verticle and horizontal profiles serve to indicate that a more complete survey would be necessary to establish accurate efficiencies.

The maximum efficiencies of 112% reported for the 1" cane are known to be at least 16% high, but a complete exhaust area survey could easily account for this amount of error.

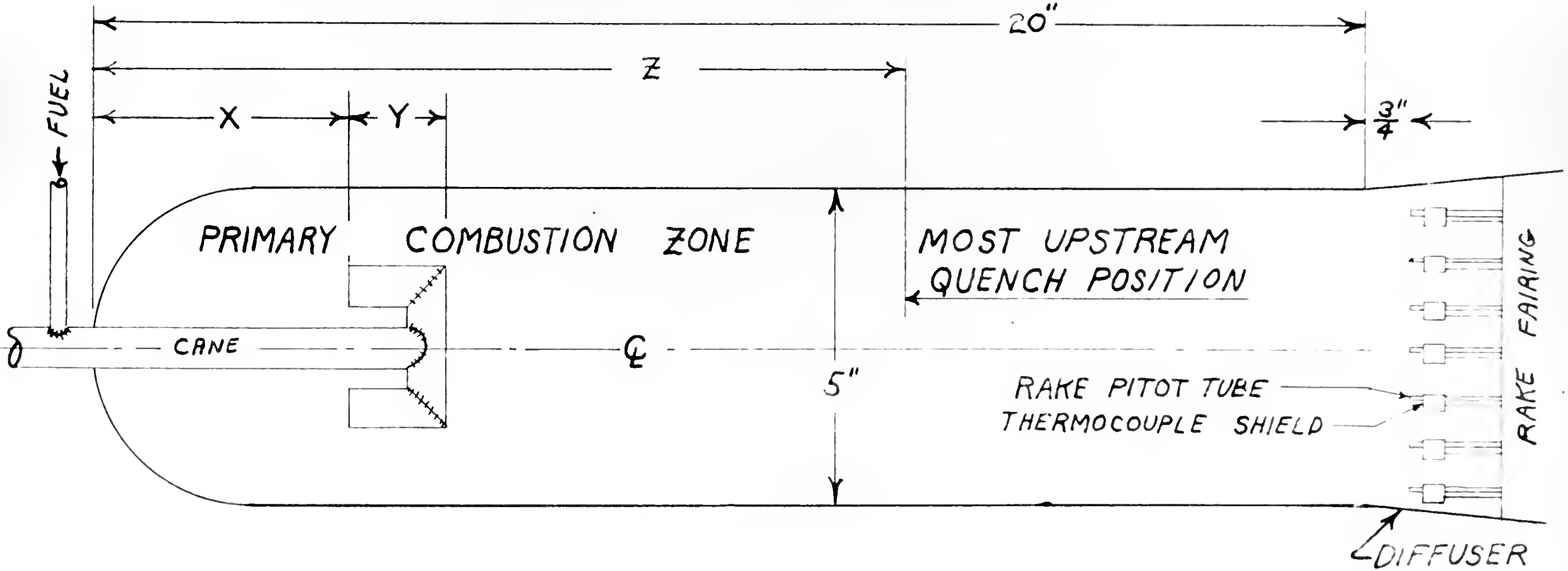


FIG. 1. SIDE ELEVATION OF COMBUSTOR SHOWING CANE AND RAKE LOCATIONS

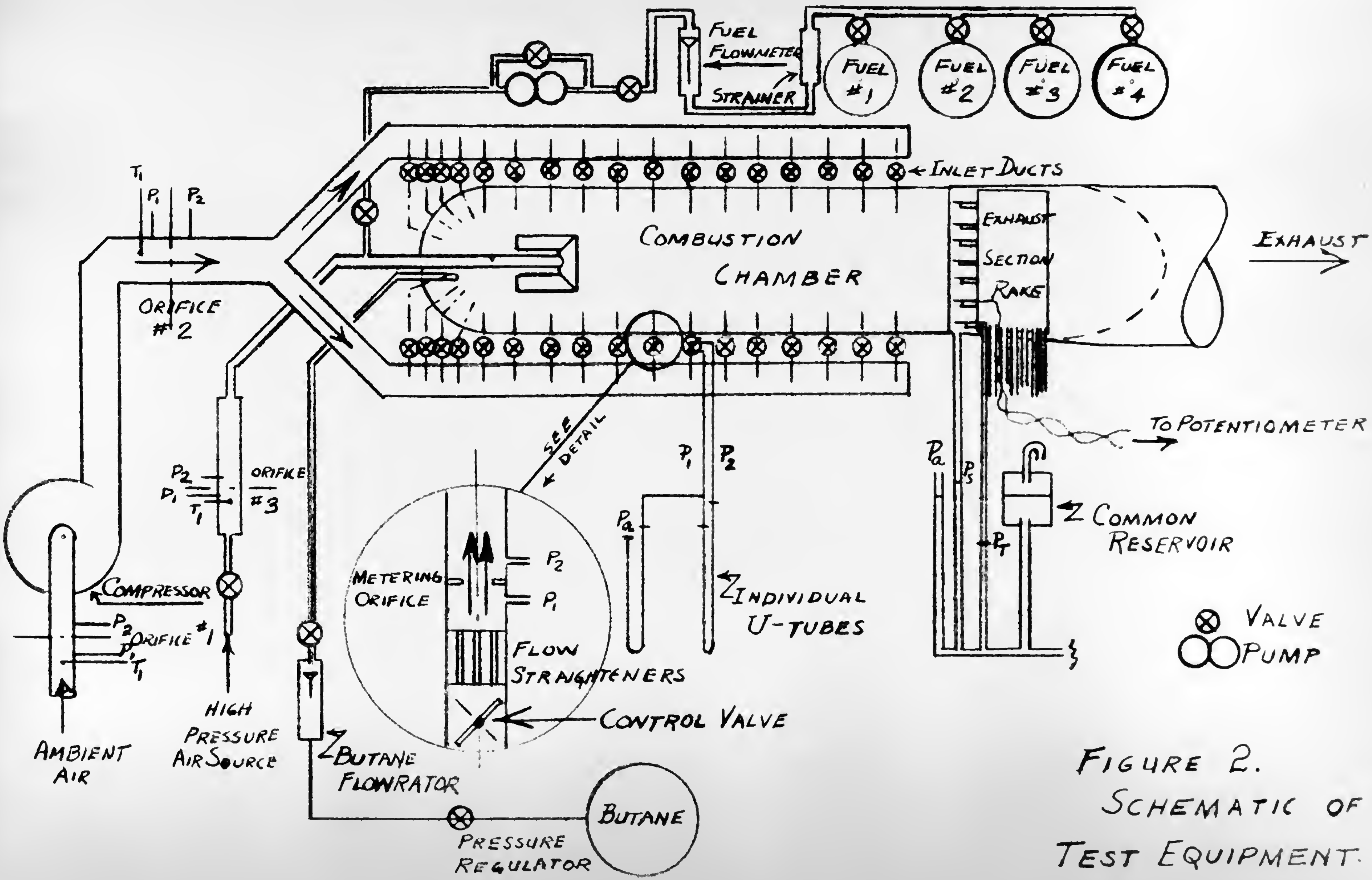


Figure 2.
Schematic of
Test Equipment.

Fig. 3 Combustor with Side Plate Installed

Fig. 4 Combustor with Side Plate Removed

Fig. 5 Vaporizer Tube (T Cane) Weldment

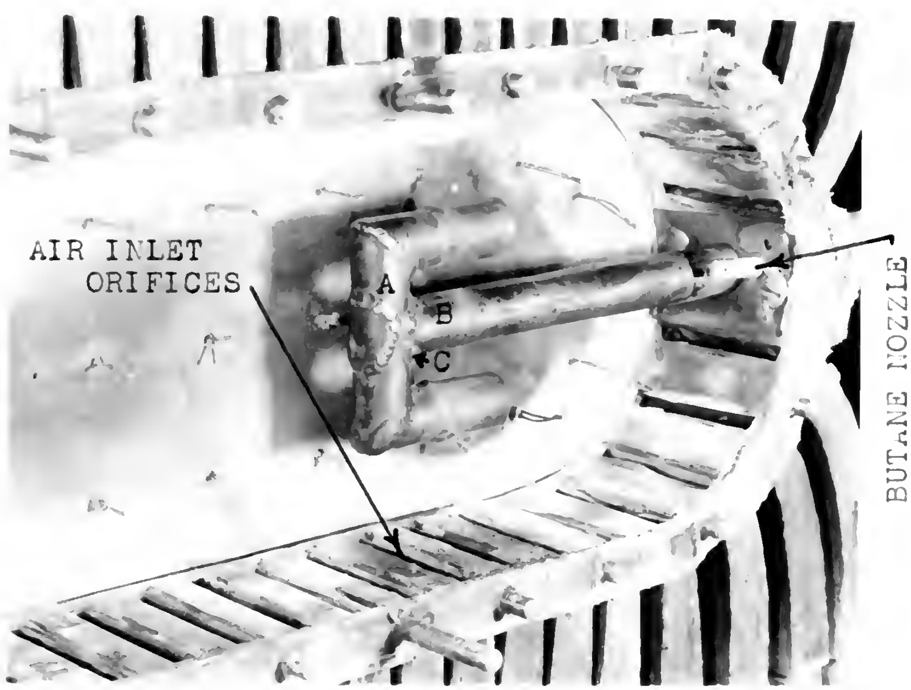


Fig. 6 Vaporizer Tube Installed

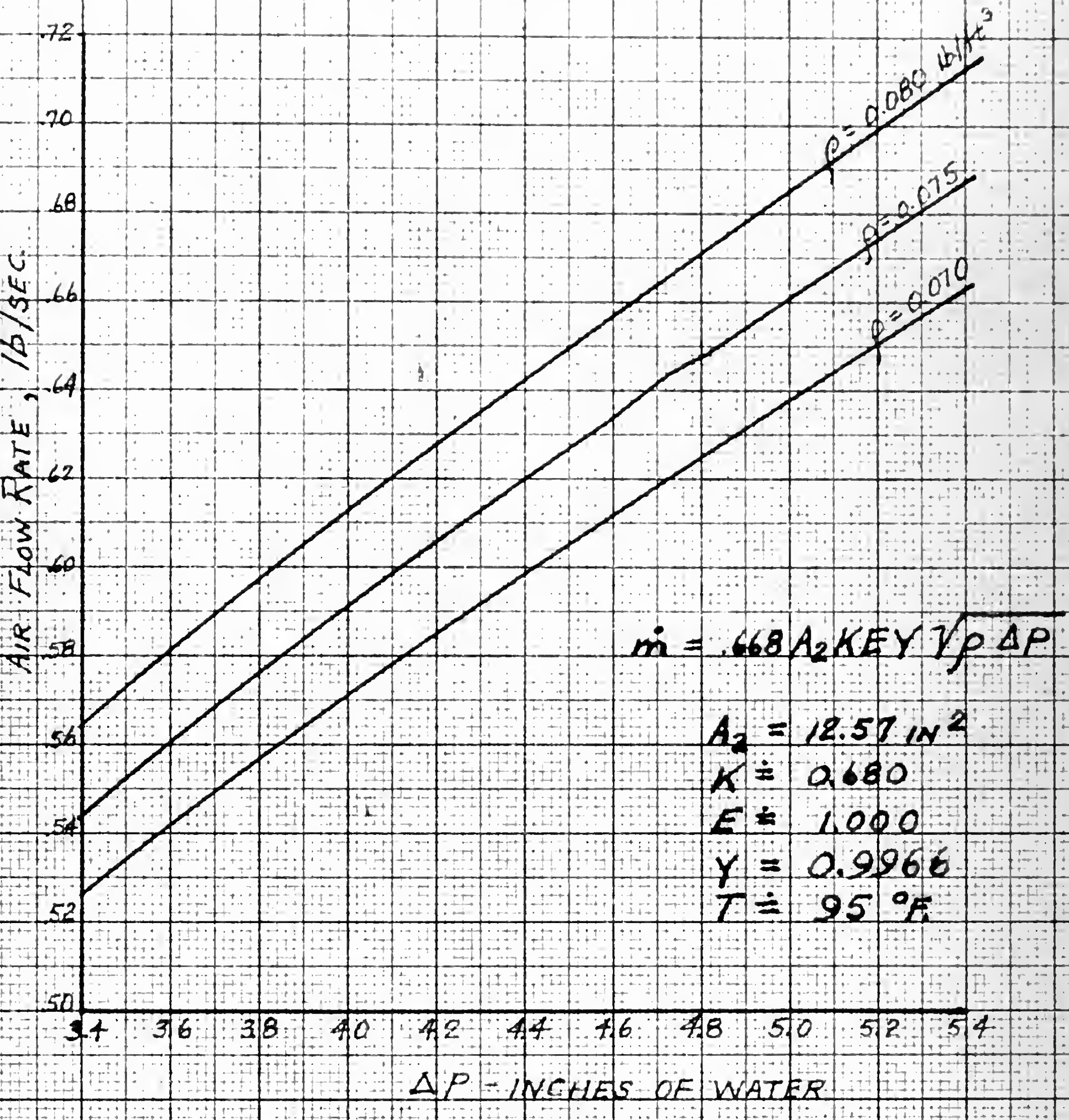


FIGURE 7. - AIR FLOW RATE FOR ORIFICE #2.

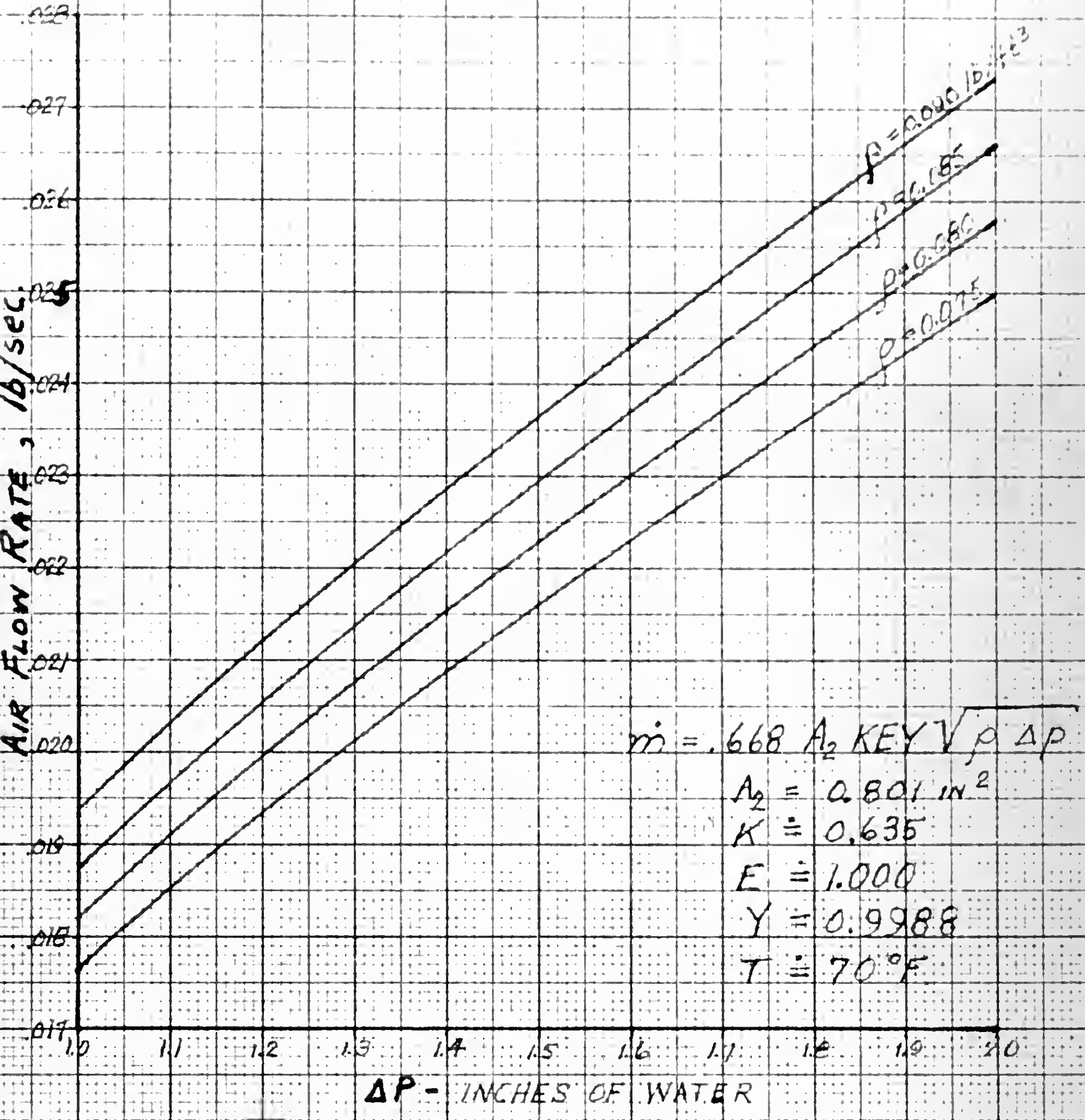


FIGURE 8. - AIR FLOW RATE FOR ORIFICE # 3

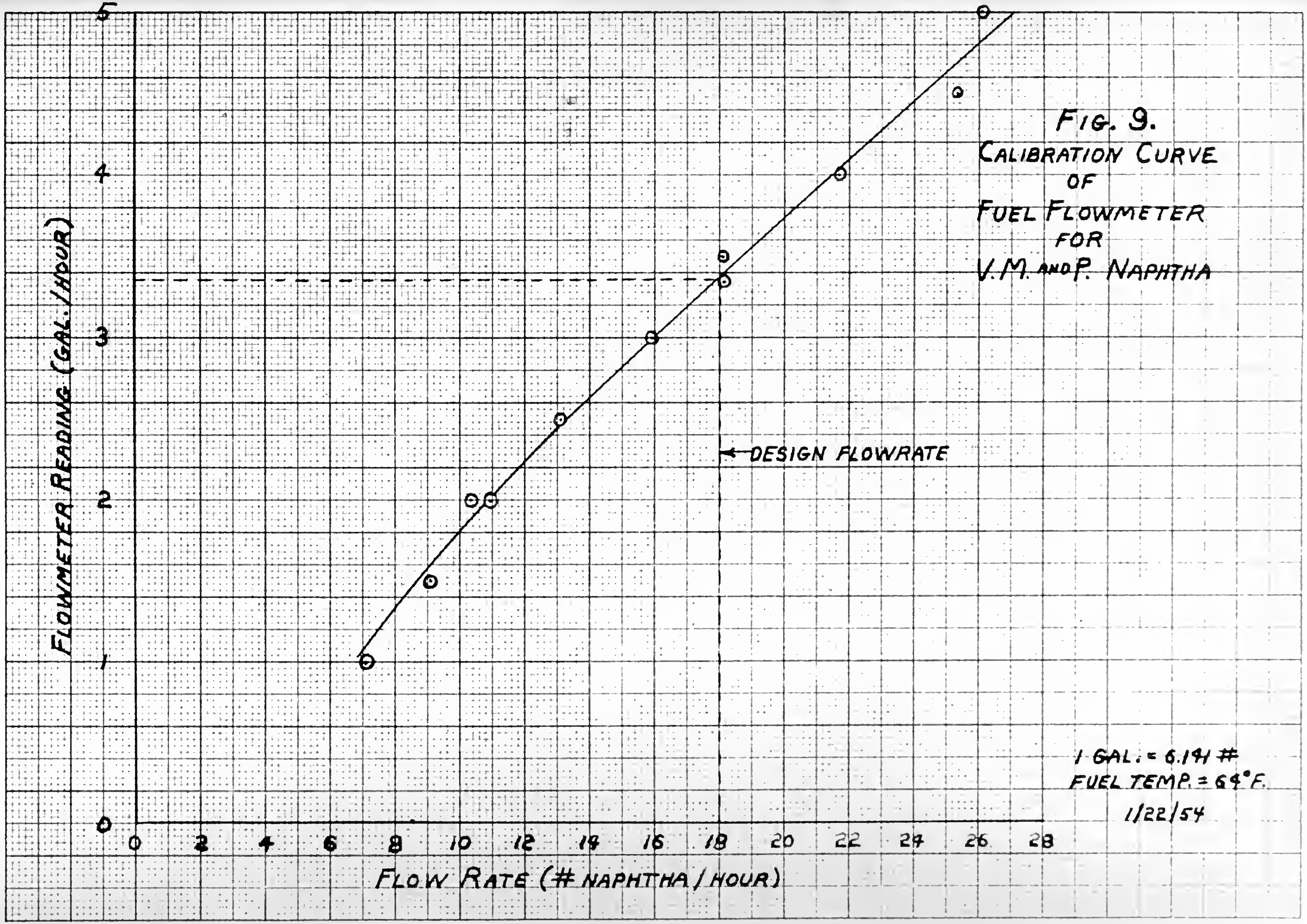


FIG. 9.
CALIBRATION CURVE
OF
FUEL FLOWMETER
FOR
V.M. AND P. NAPHTHA
←DESIGN FLOWRATE
1 GAL. = 6.191 #
FUEL TEMP. = 64°F.
1/22/54
FLOWMETER READING (GAL./HOUR)
FLOW RATE (# NAPHTHA / HOUR)
0
1
2
3
4
5
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28

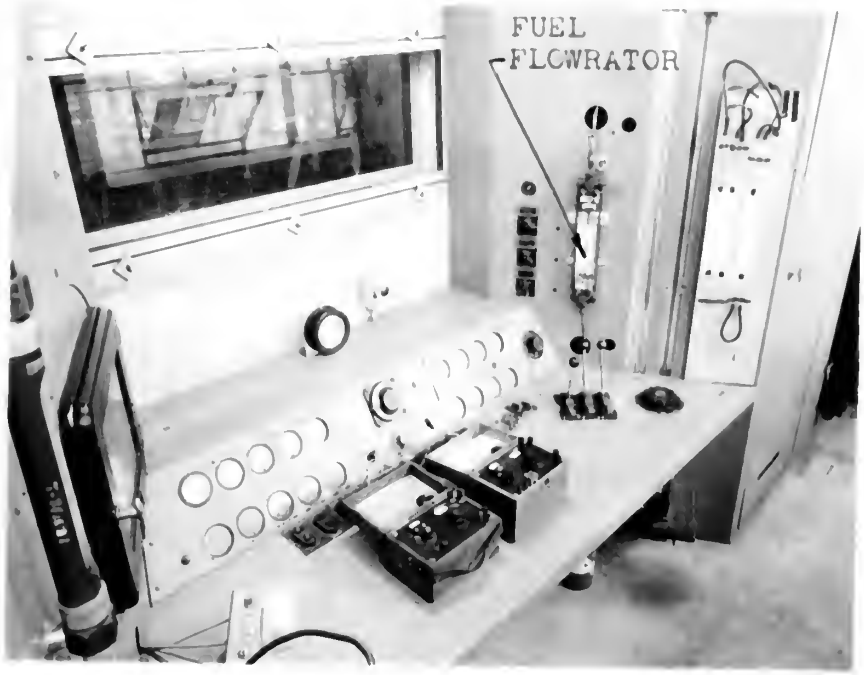


Fig. 11 Control Room

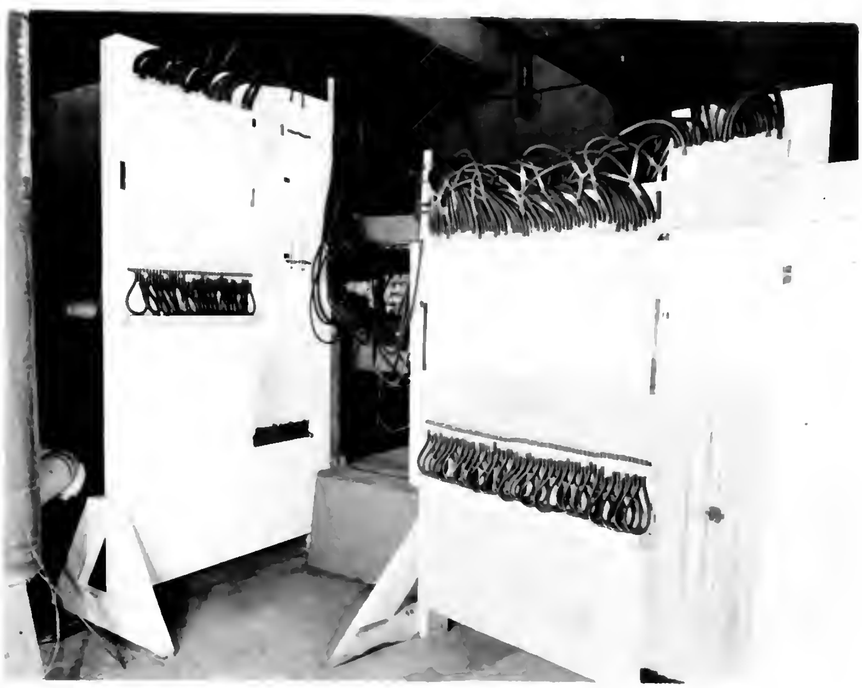


Fig. 10 Manometer Assemblies

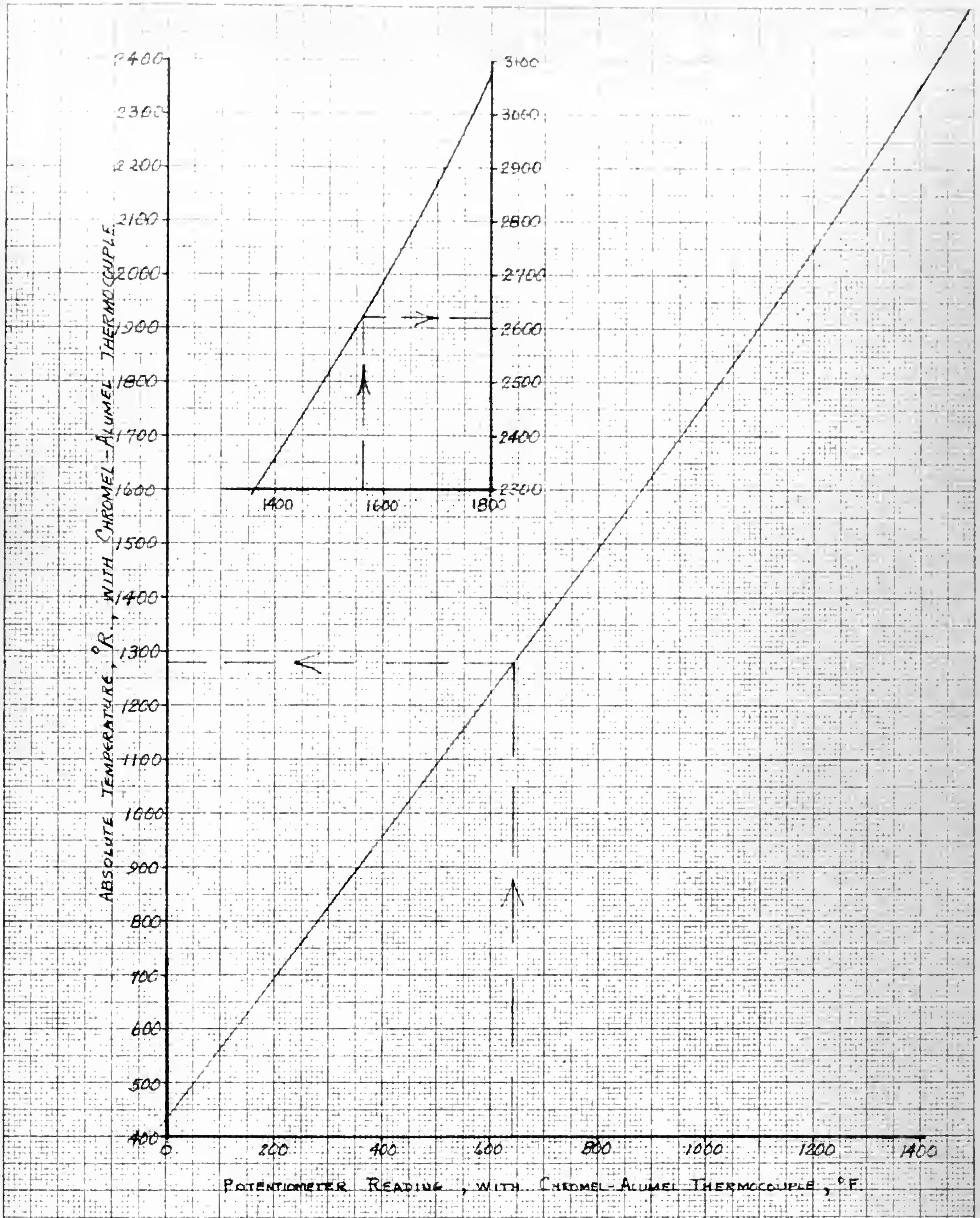


Figure 12 Thermocouple Conversion Curve.

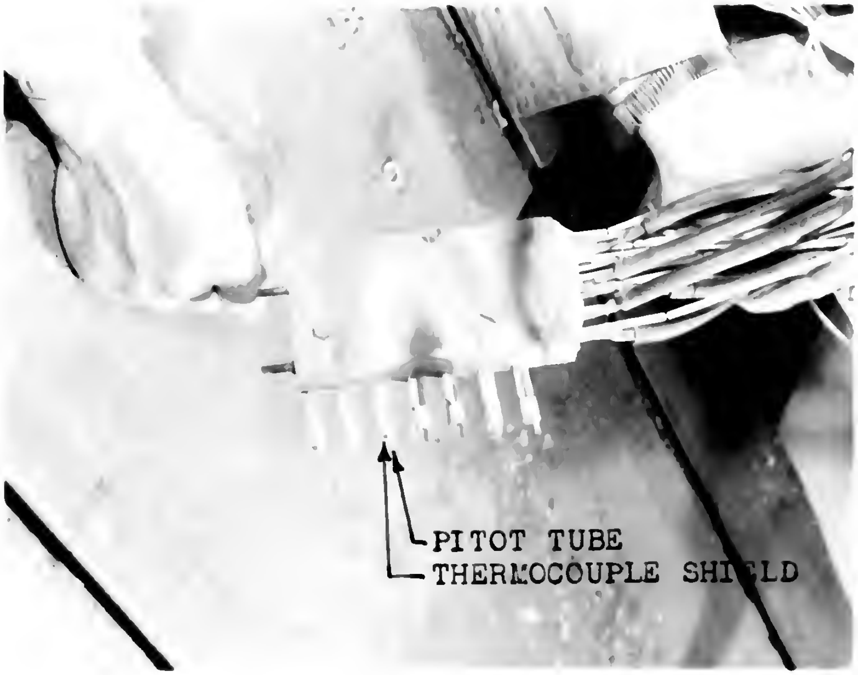


Fig. 14 Exhaust Rake Assembled

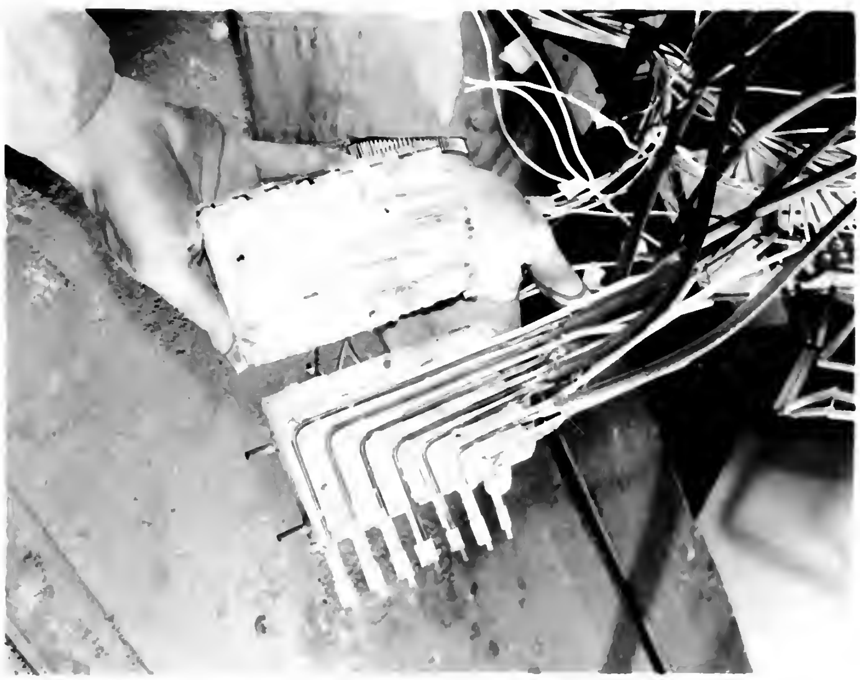


Fig. 13 Exhaust Rake Split at Aluminum Foil

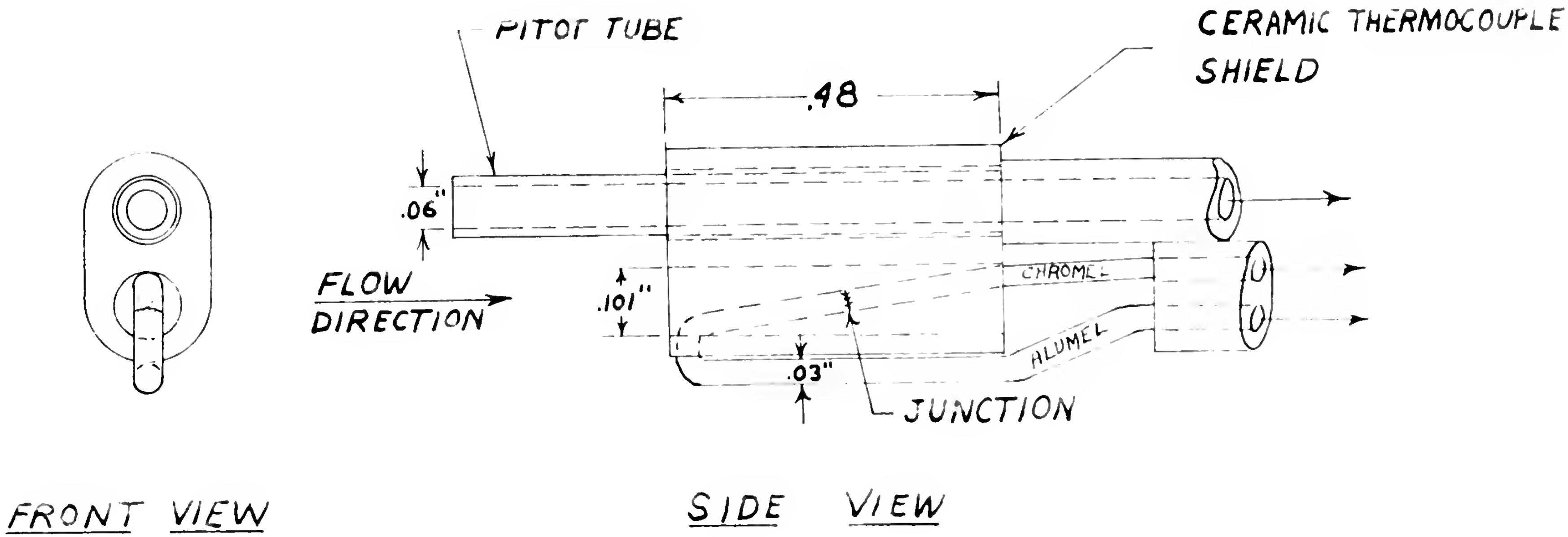
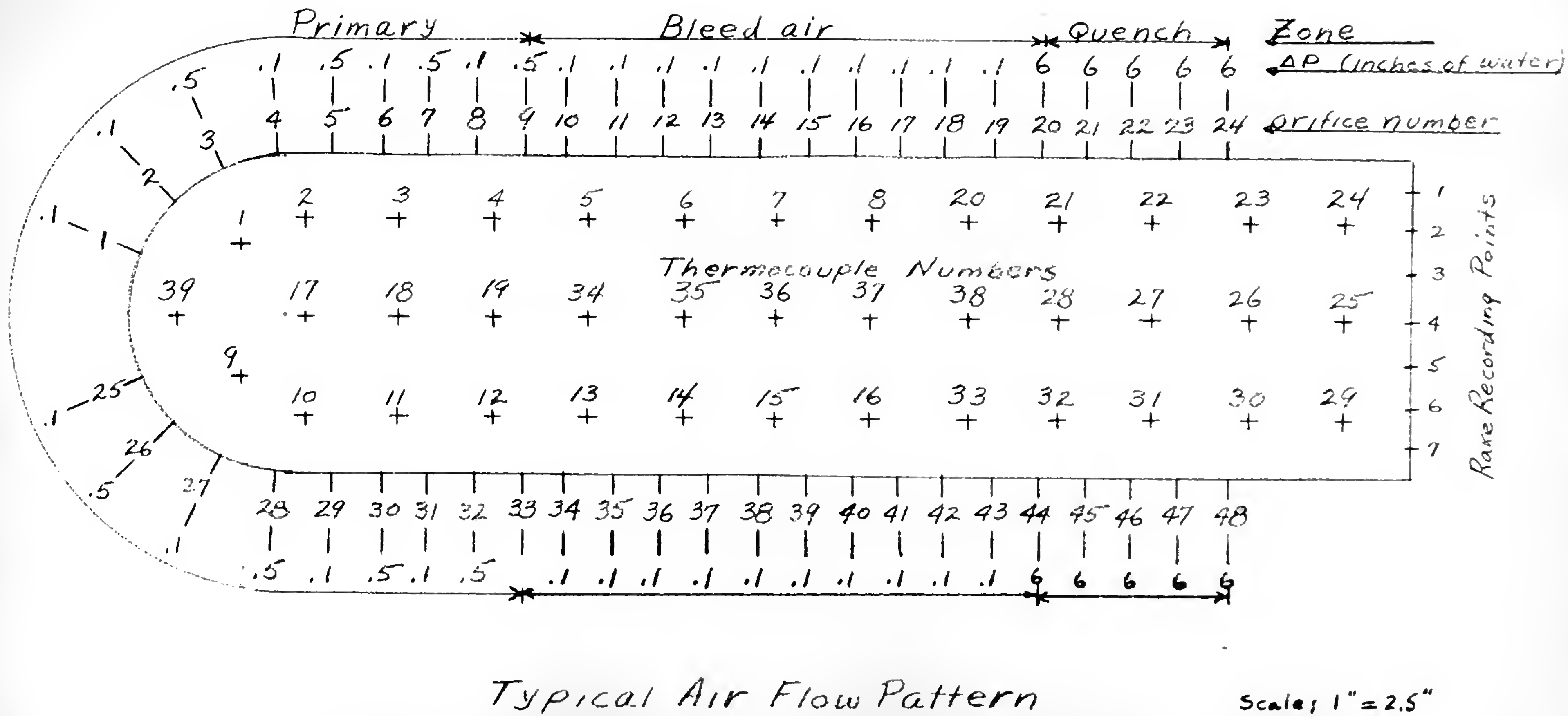


FIG. 15. RAKE THERMOCOUPLE AND SHIELD DETAIL



Typical Air Flow Pattern

Z- Position ~ 20

FIG. 16.

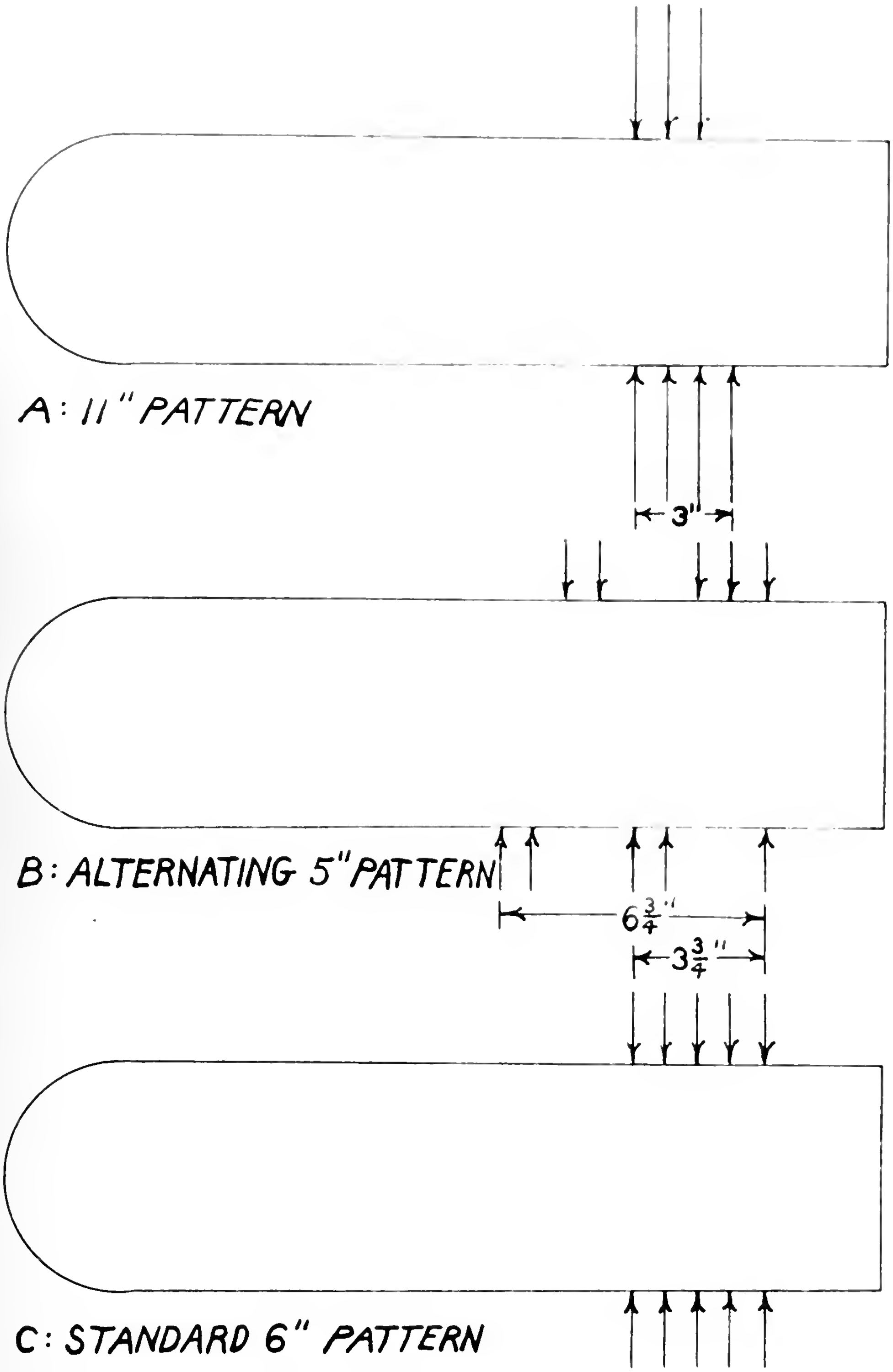


FIG. 17 : QUENCH AIR PATTERNS TESTED

FIG 18
BUTANE
2" CANE

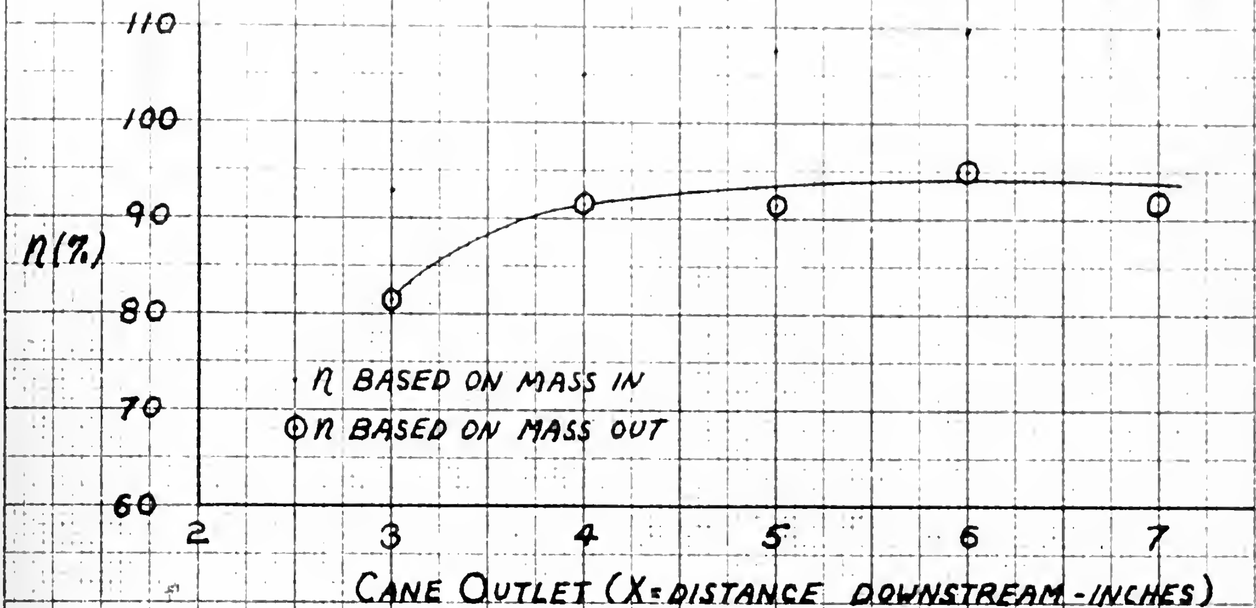
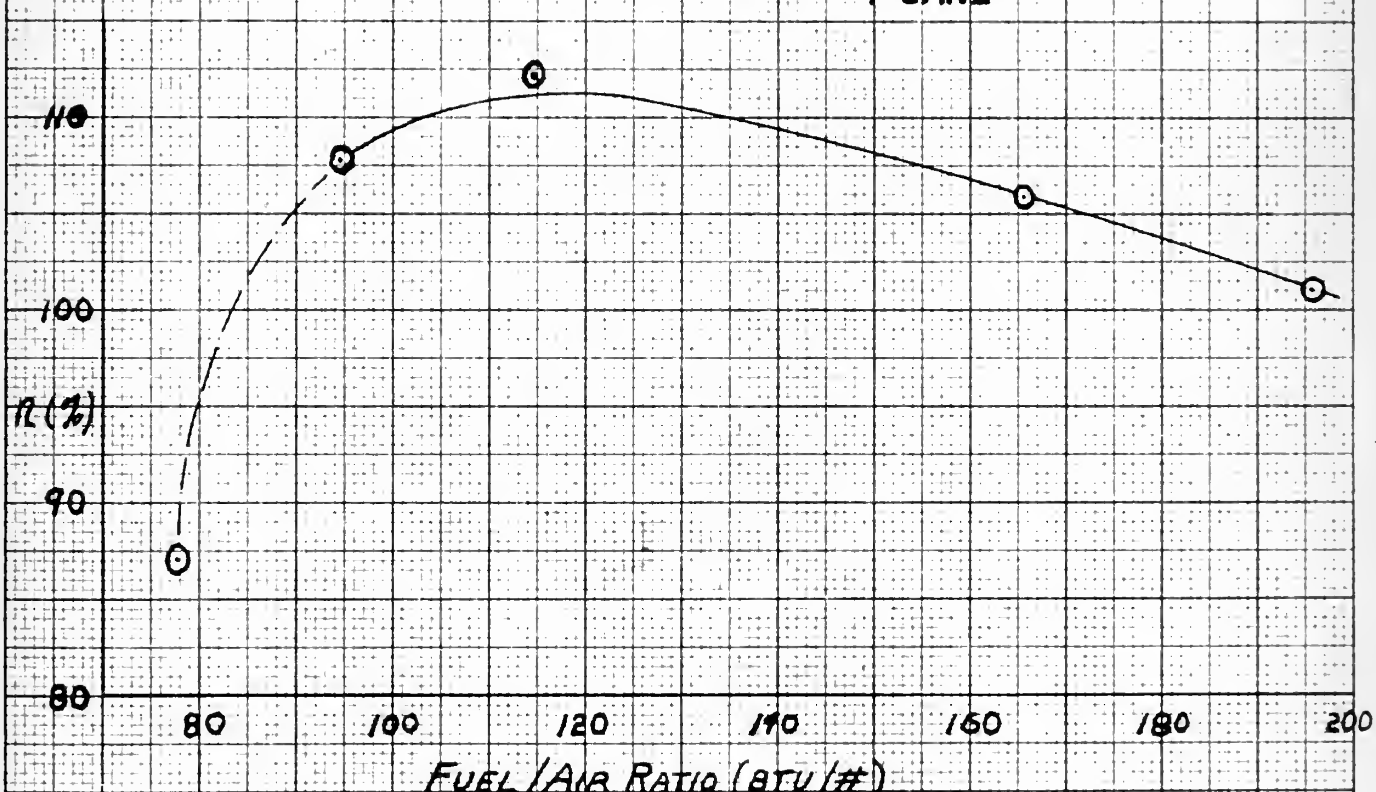


FIG 19
NAPHTHA
1" CANE



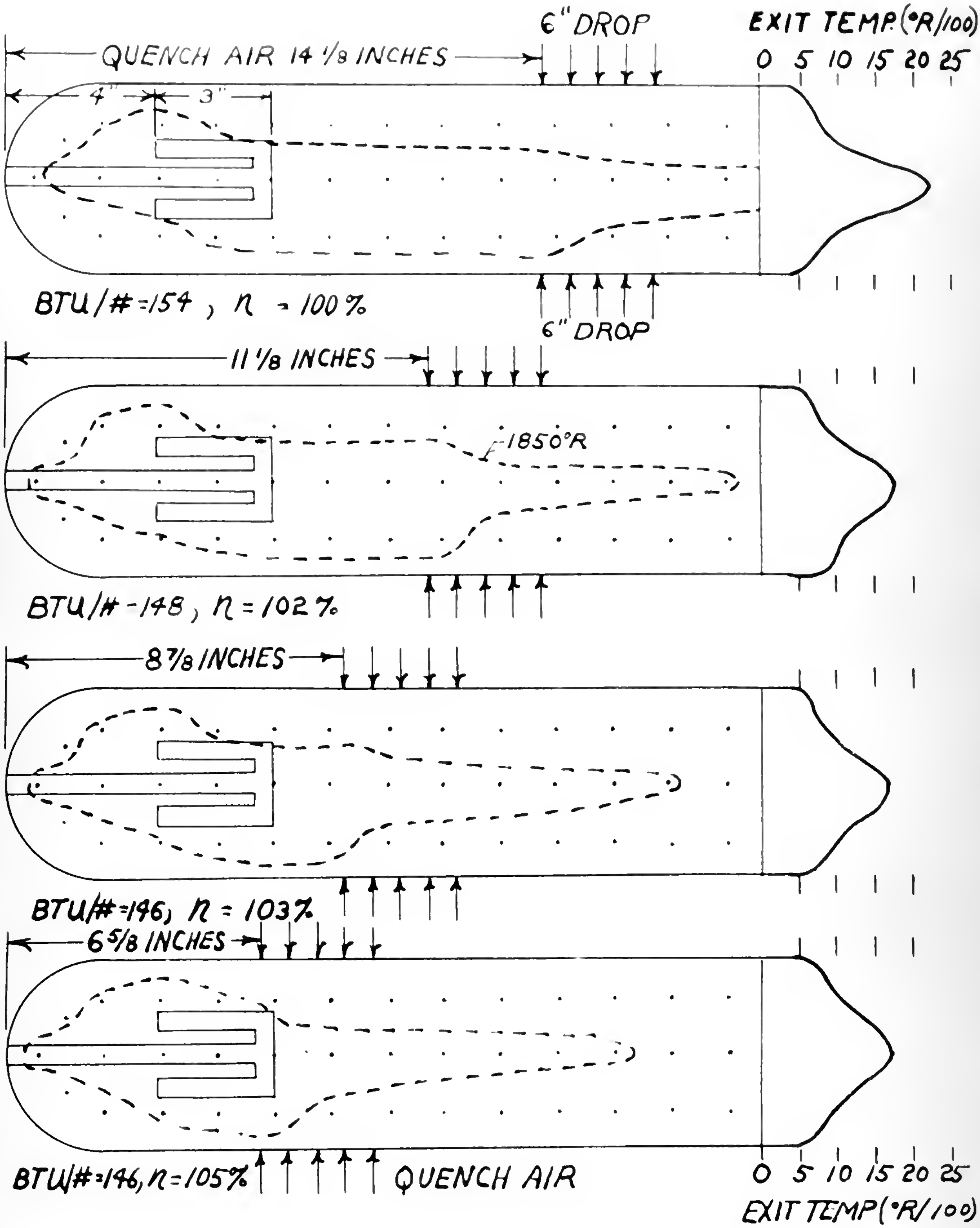


FIG. 20 VARIATION OF COMBUSTION AREA, EFFICIENCY, AND TEMPERATURE PROFILE WITH QUENCH AIR POSITION (FUEL/AIR RATIO - 148 BTU/#)

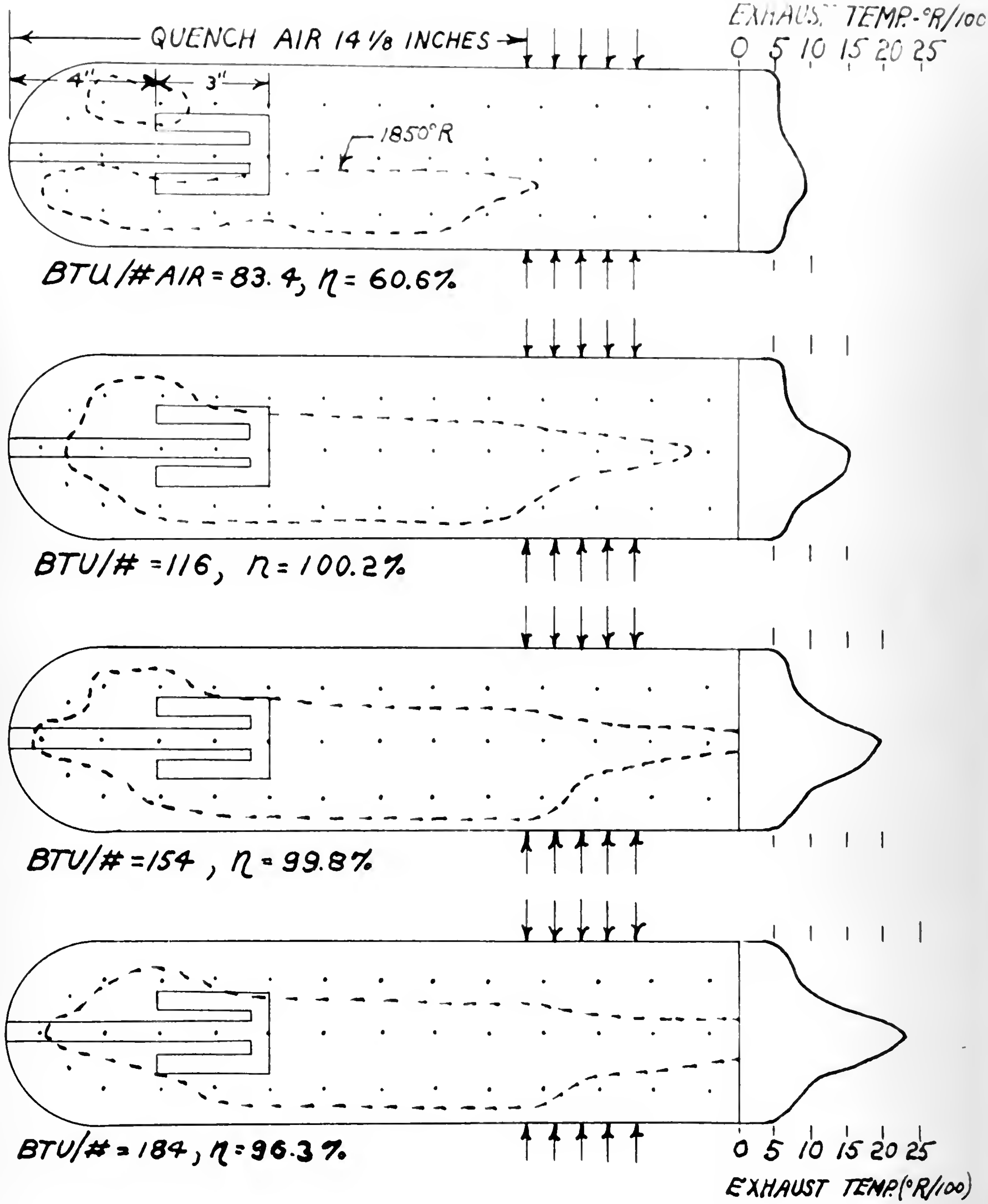


FIG. 21 VARIATION OF COMBUSTION AREA, EFFICIENCY, AND EXAUST TEMPERATURE PROFILE WITH FUEL/AIR RATIO (QUENCH AIR 14 1/8" DOWNSTREAM)

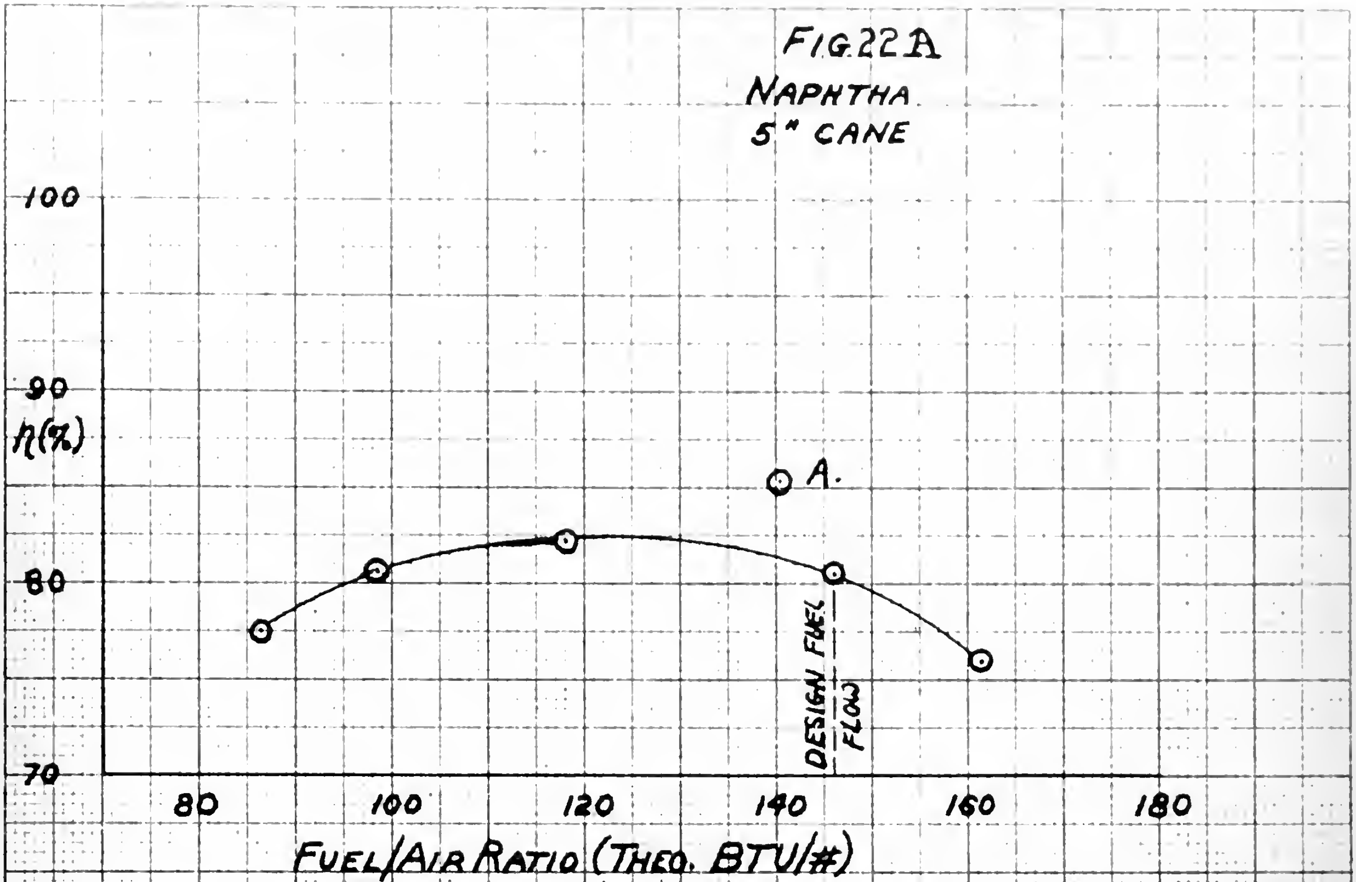


FIG. 22A
NAPHTHA
5" CANE
100
90
η(%)
80
70
A.
DESIGN FUEL FLOW
80
100
120
140
160
180
FUEL/AIR RATIO (THEO. BTU/#)

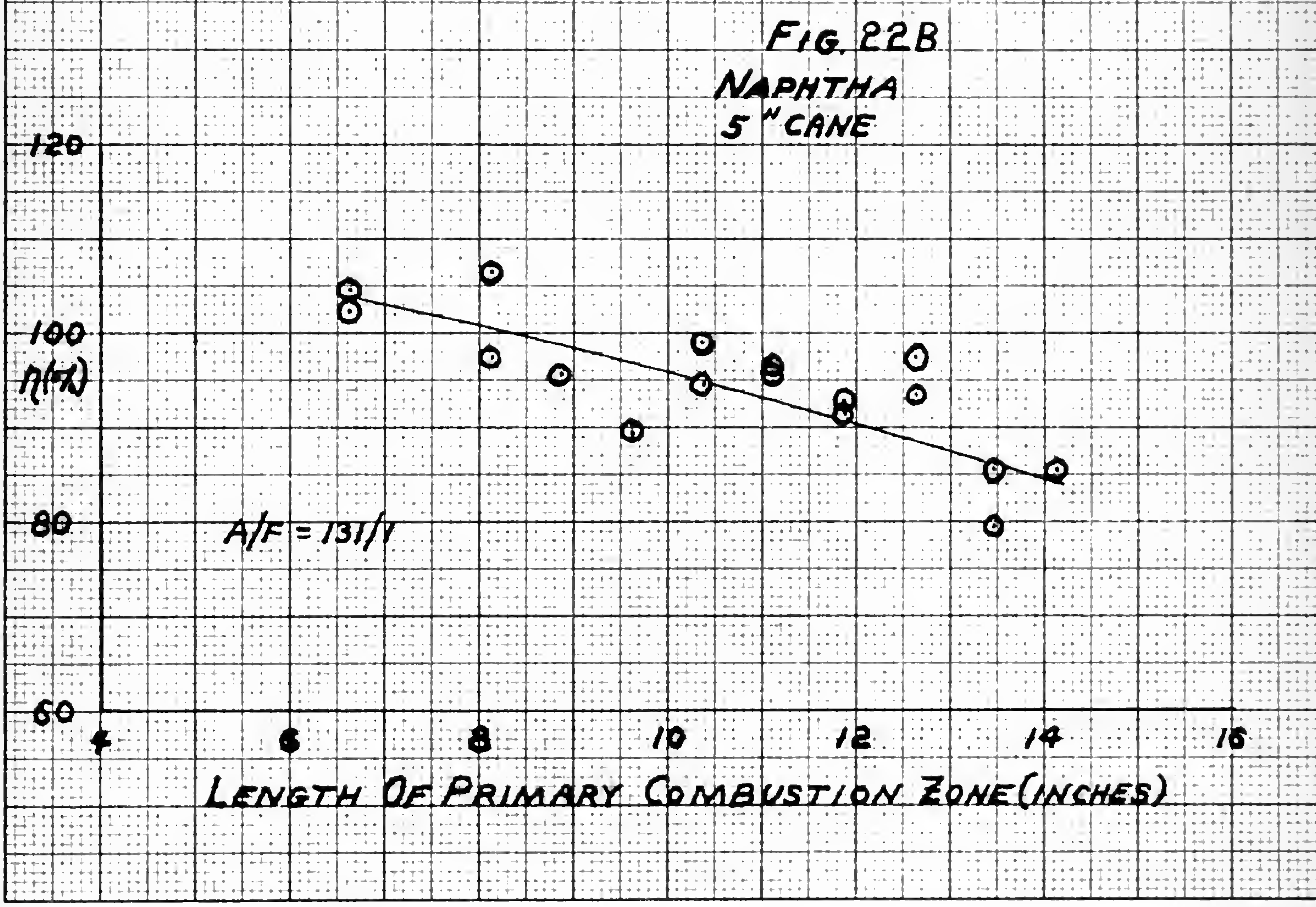


FIG. 22B
NAPHTHA
5" CANE
120
100
η(%)
80
60
A/F = 131/1
4
6
8
10
12
14
16
LENGTH OF PRIMARY COMBUSTION ZONE (INCHES)

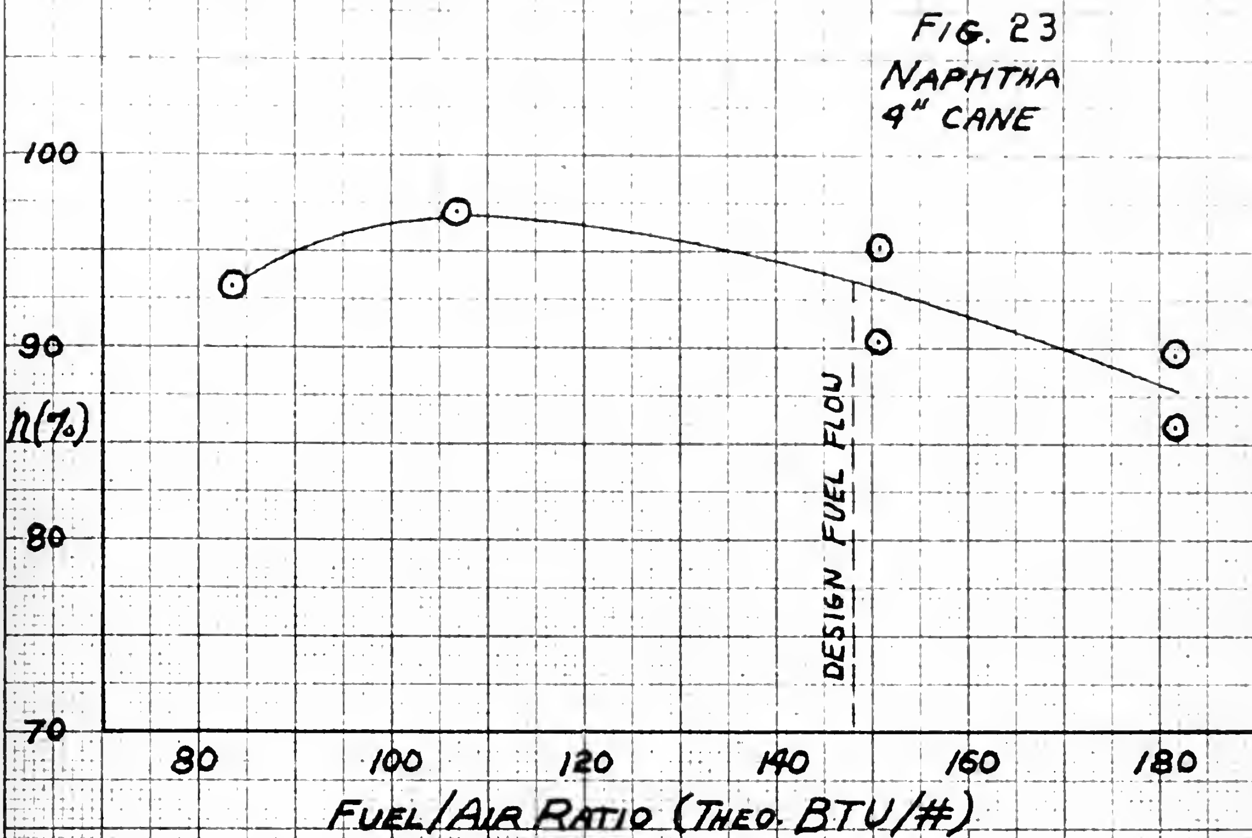


FIG. 23
NAPHTHA
4" CANE
100
90
η(%)
80
70
DESIGN FUEL FLOW
80
100
120
140
160
180
FUEL/AIR RATIO (THEO. BTU/#)

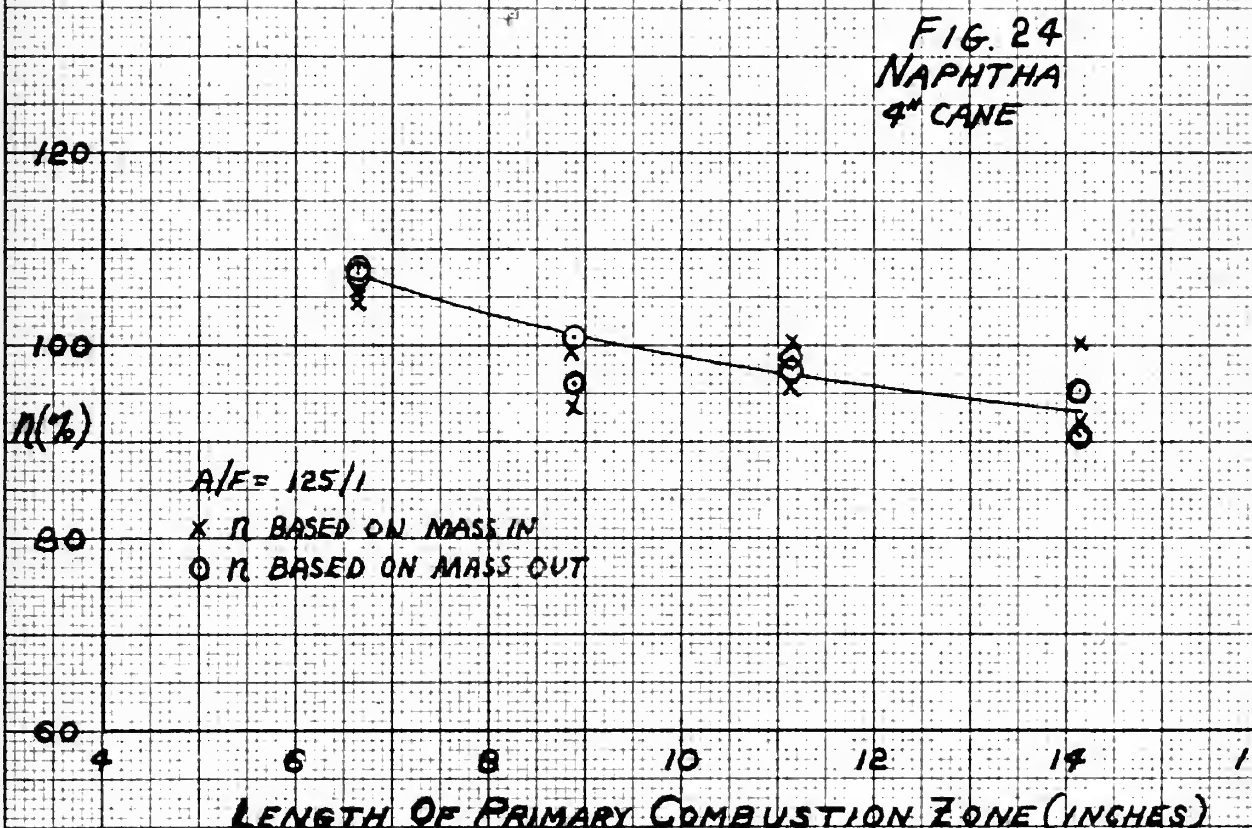


FIG. 24
NAPHTHA
4" CANE
120
100
η(%)
80
60
A/F = 125/1
× η BASED ON MASS IN
⊙ η BASED ON MASS OUT
4
6
8
10
12
14
16
LENGTH OF PRIMARY COMBUSTION ZONE (INCHES)

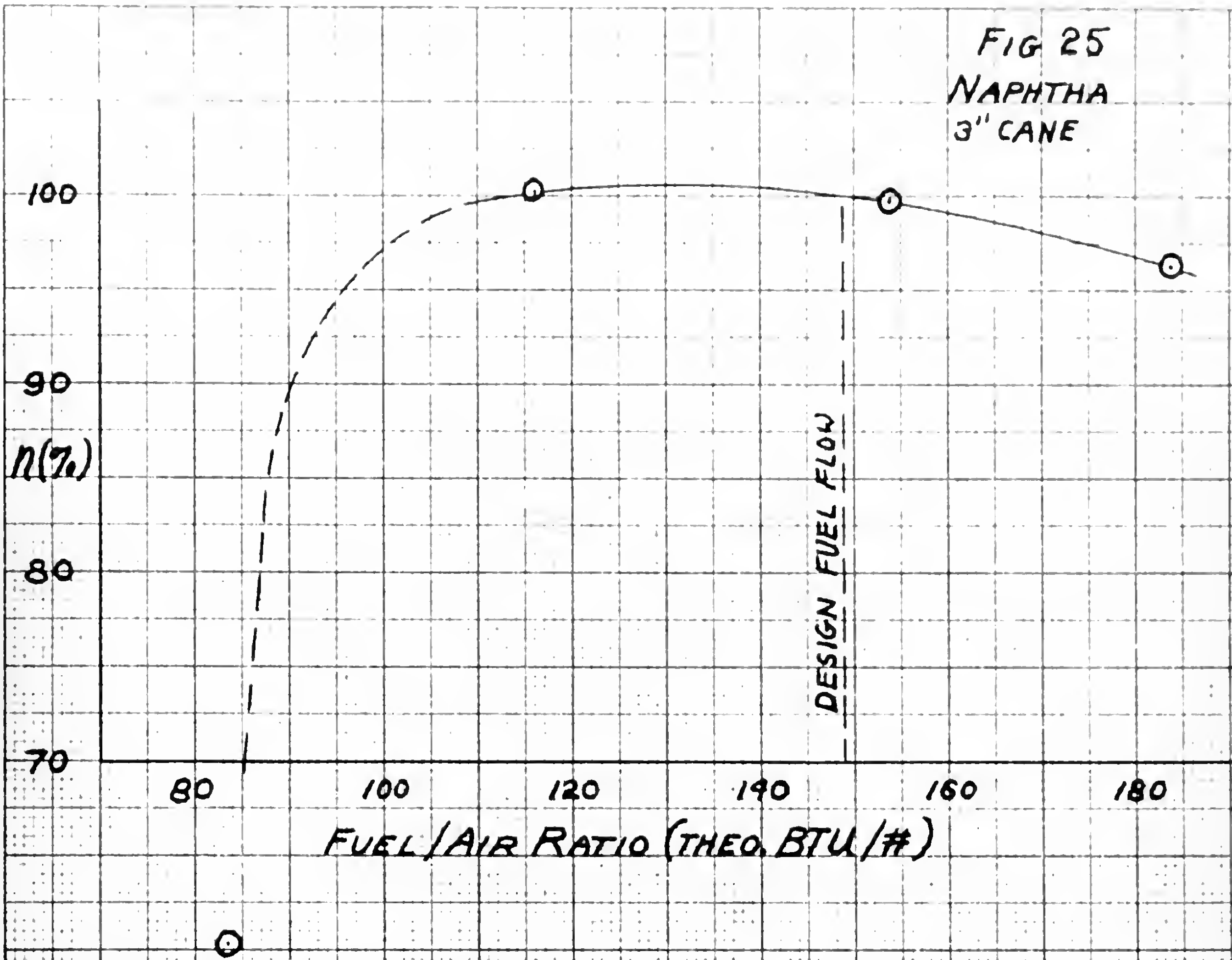


FIG 25
NAPHTHA
3" CANE
100
90
η(%)
80
70
80
100
120
140
160
180
DESIGN FUEL FLOW
FUEL/AIR RATIO (THEO. BTU/#)

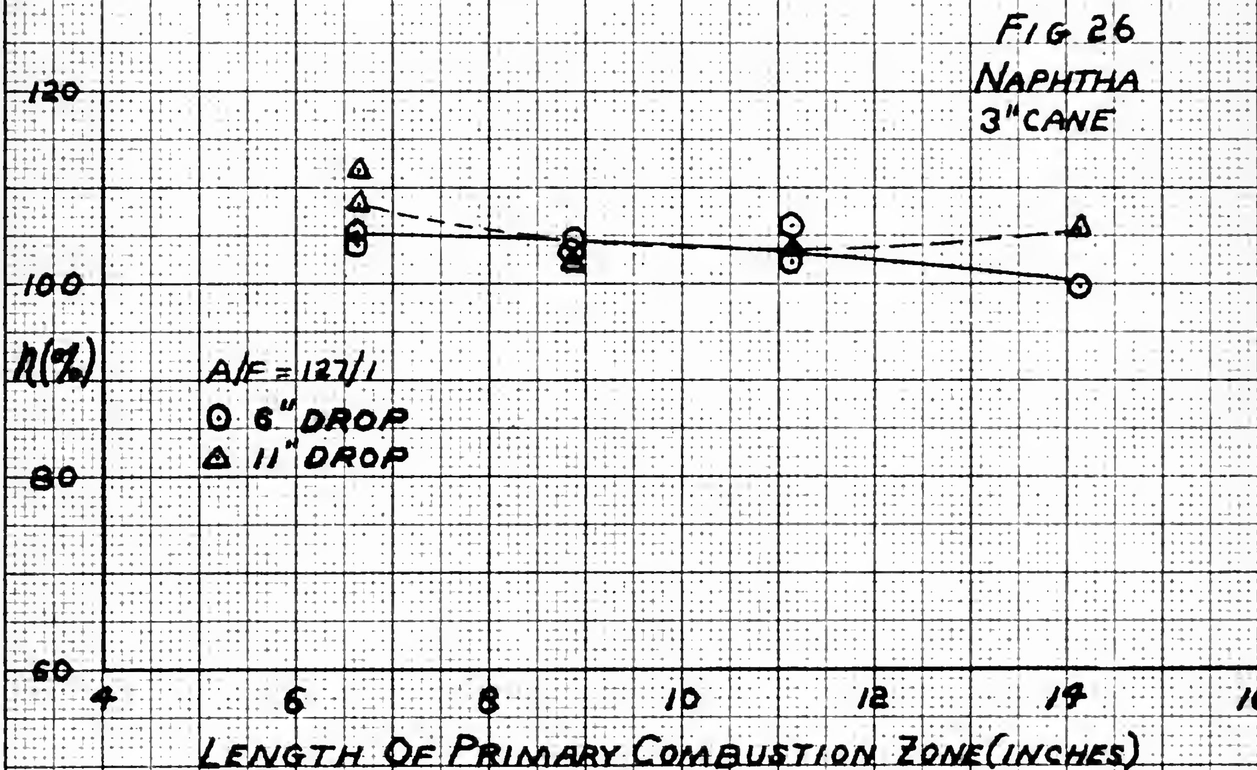


FIG 26
NAPHTHA
3" CANE
120
100
η(%)
80
60
A/F = 127/1
⊙ 6" DROP
Δ 11" DROP
4
6
8
10
12
14
16
LENGTH OF PRIMARY COMBUSTION ZONE (INCHES)

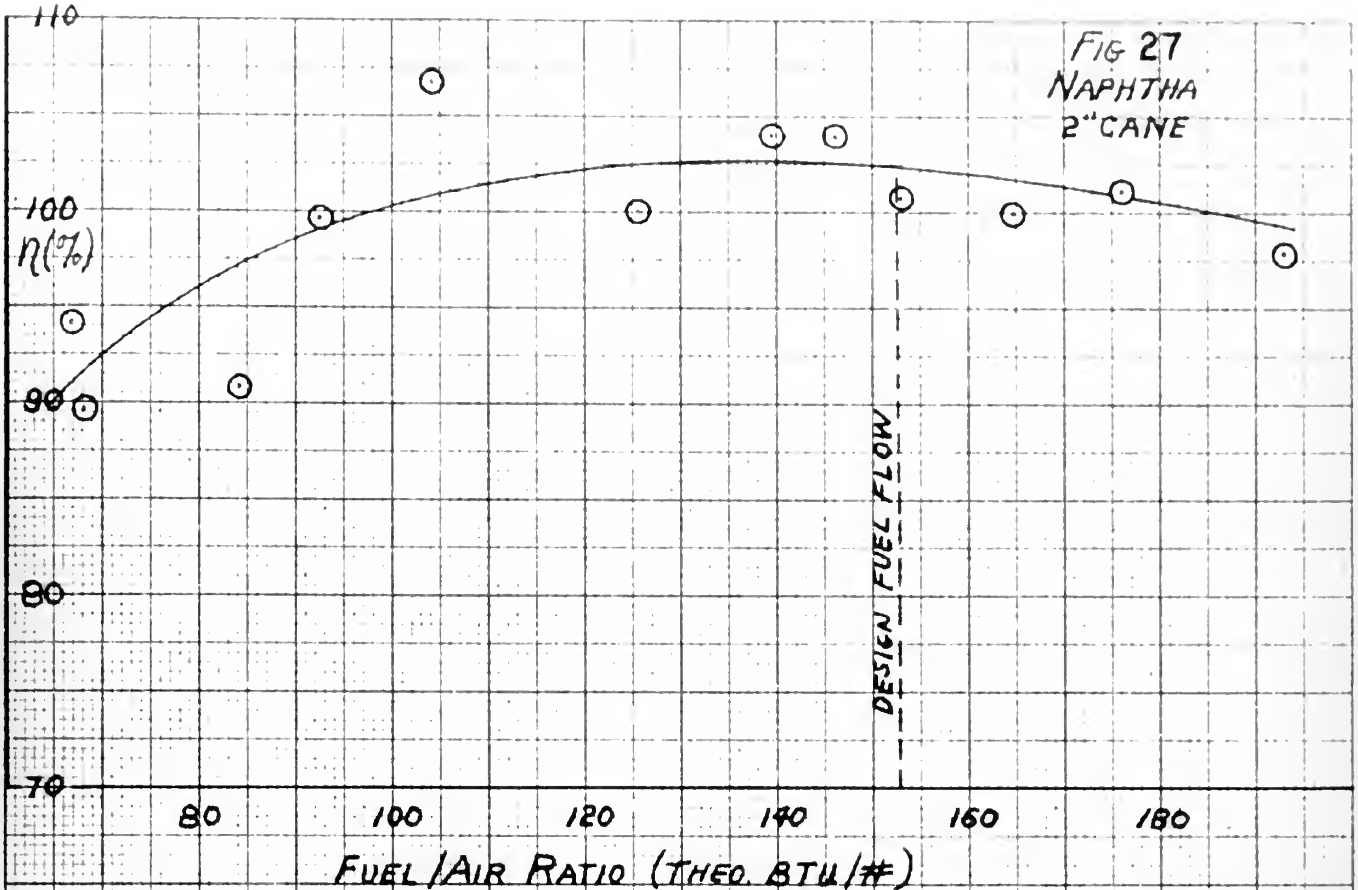


Fig 27
Naphtha
2" Cane
110
100
η(%)
90
80
70
Design Fuel Flow
80
100
120
140
160
180
Fuel/Air Ratio (Theo. BTU/#)

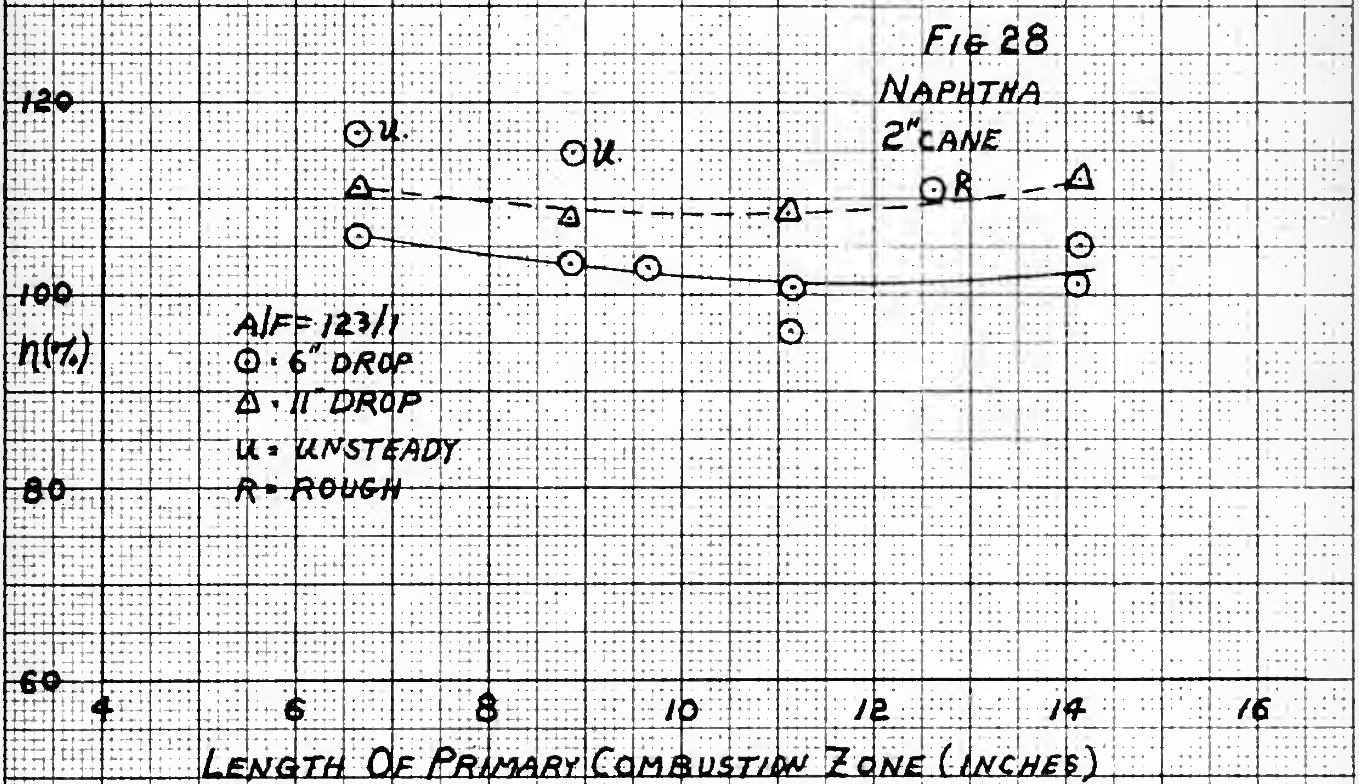


Fig 28
Naphtha
2" Cane
120
100
η(%)
80
60
U.
U.
R
A/F = 123/1
⊙ - 6" Drop
Δ - 11" Drop
U = Unsteady
R = Rough
4
6
8
10
12
14
16
Length Of Primary Combustion Zone (inches)

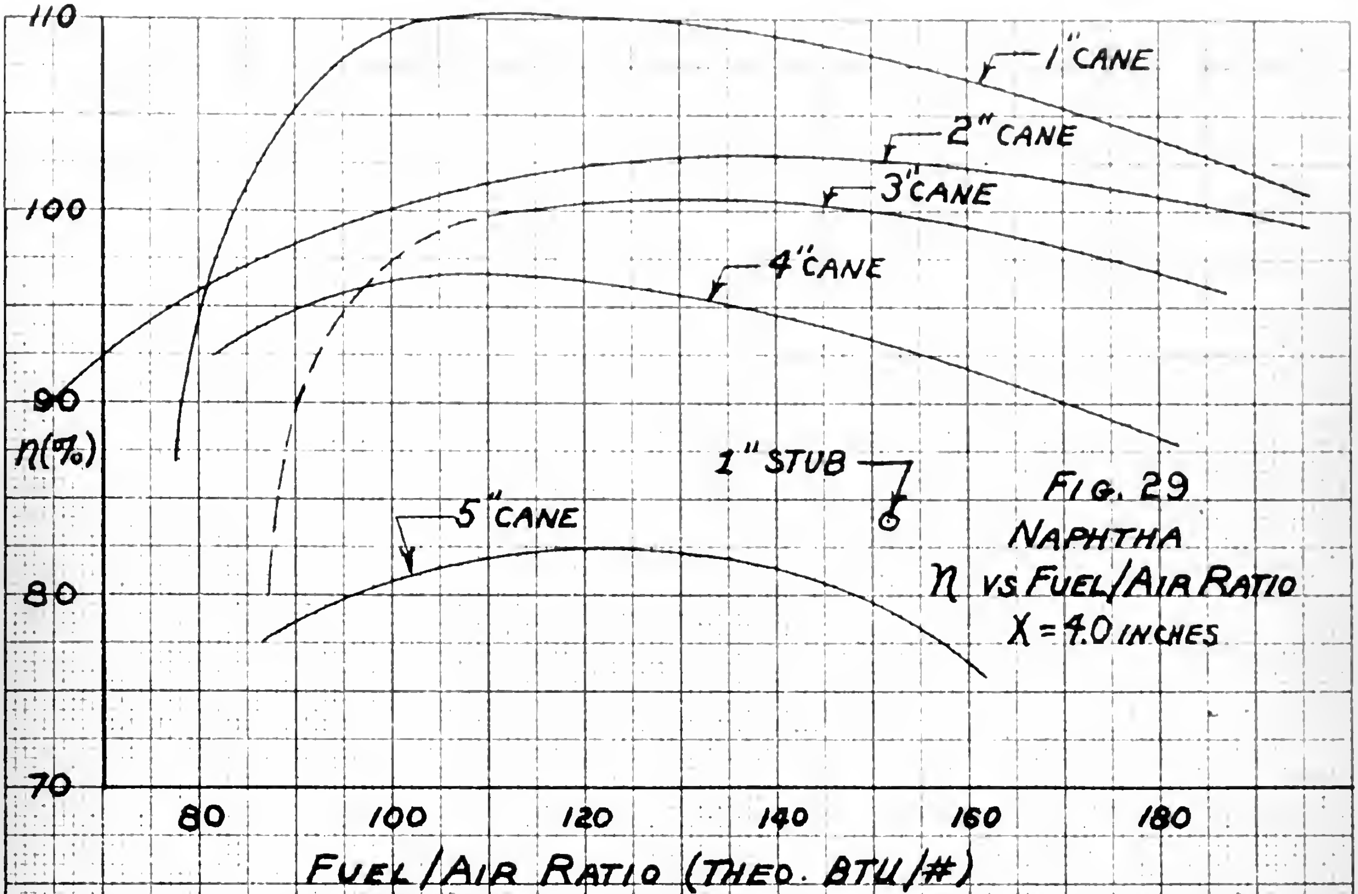


FIG. 29
NAPHTHA
η VS FUEL/AIR RATIO
X = 4.0 INCHES

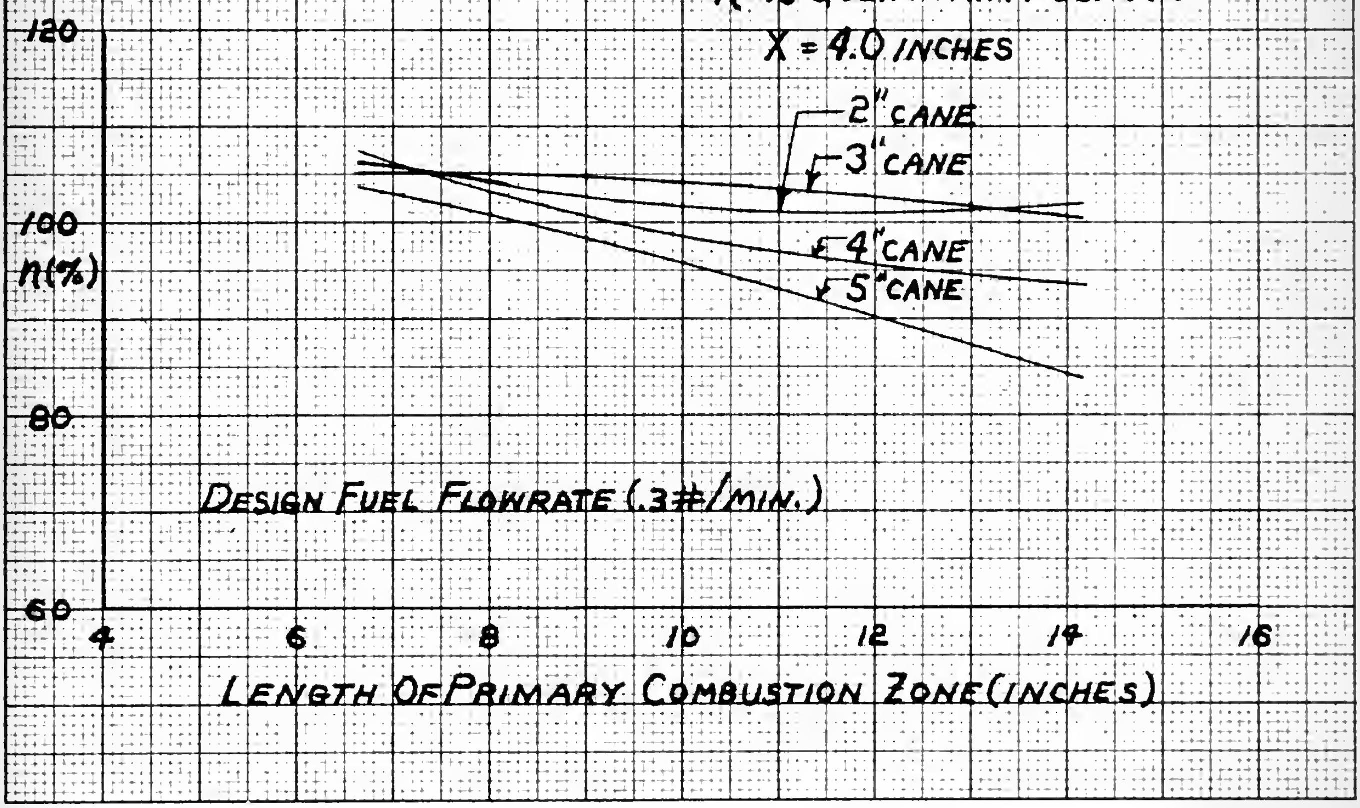
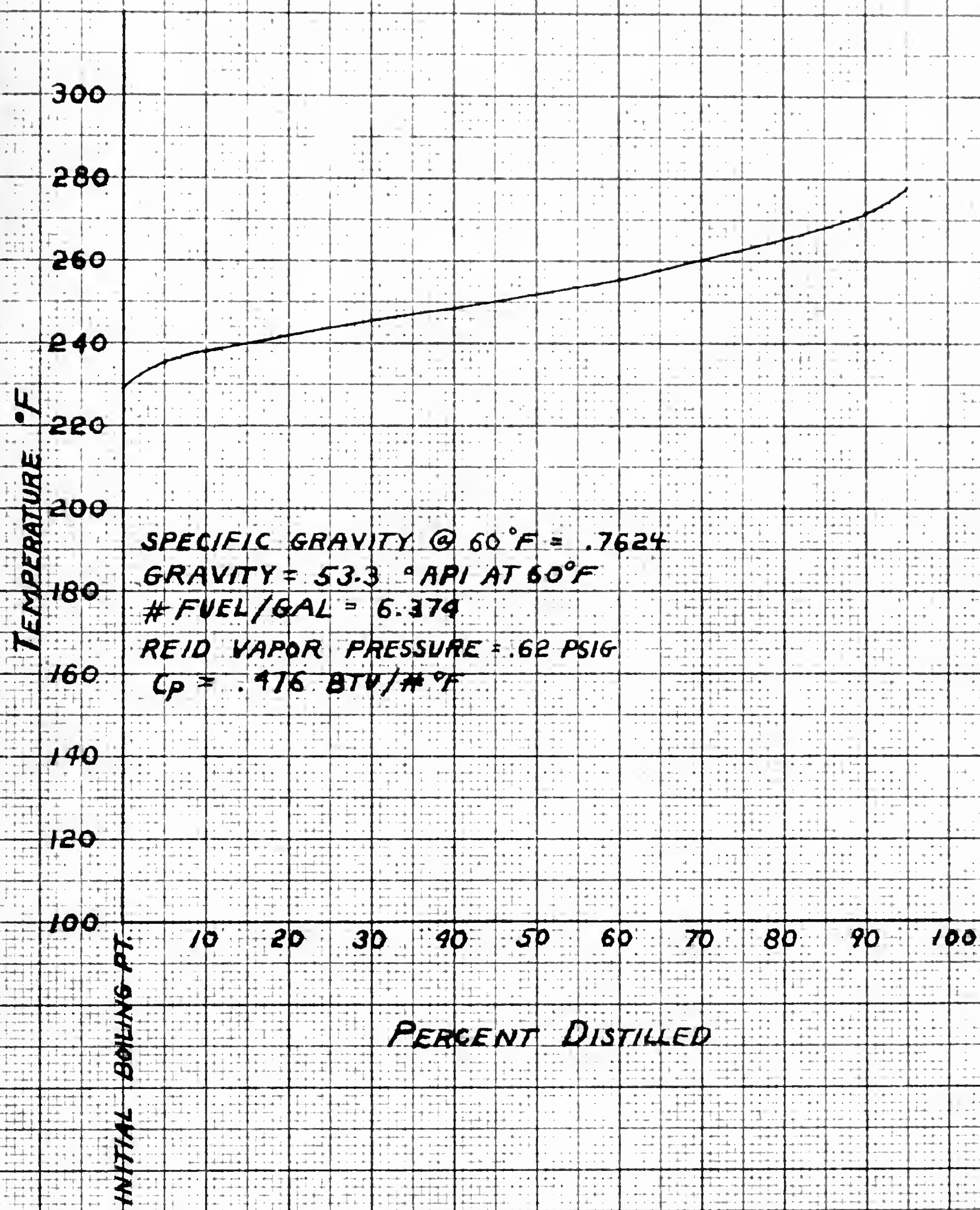


FIG 30
NAPHTHA
η VS QUENCH AIR POSITION
X = 4.0 INCHES

FIG [redacted] 31

V. M. & P. NAPHTHA ASTM DISTILLATION CURVE



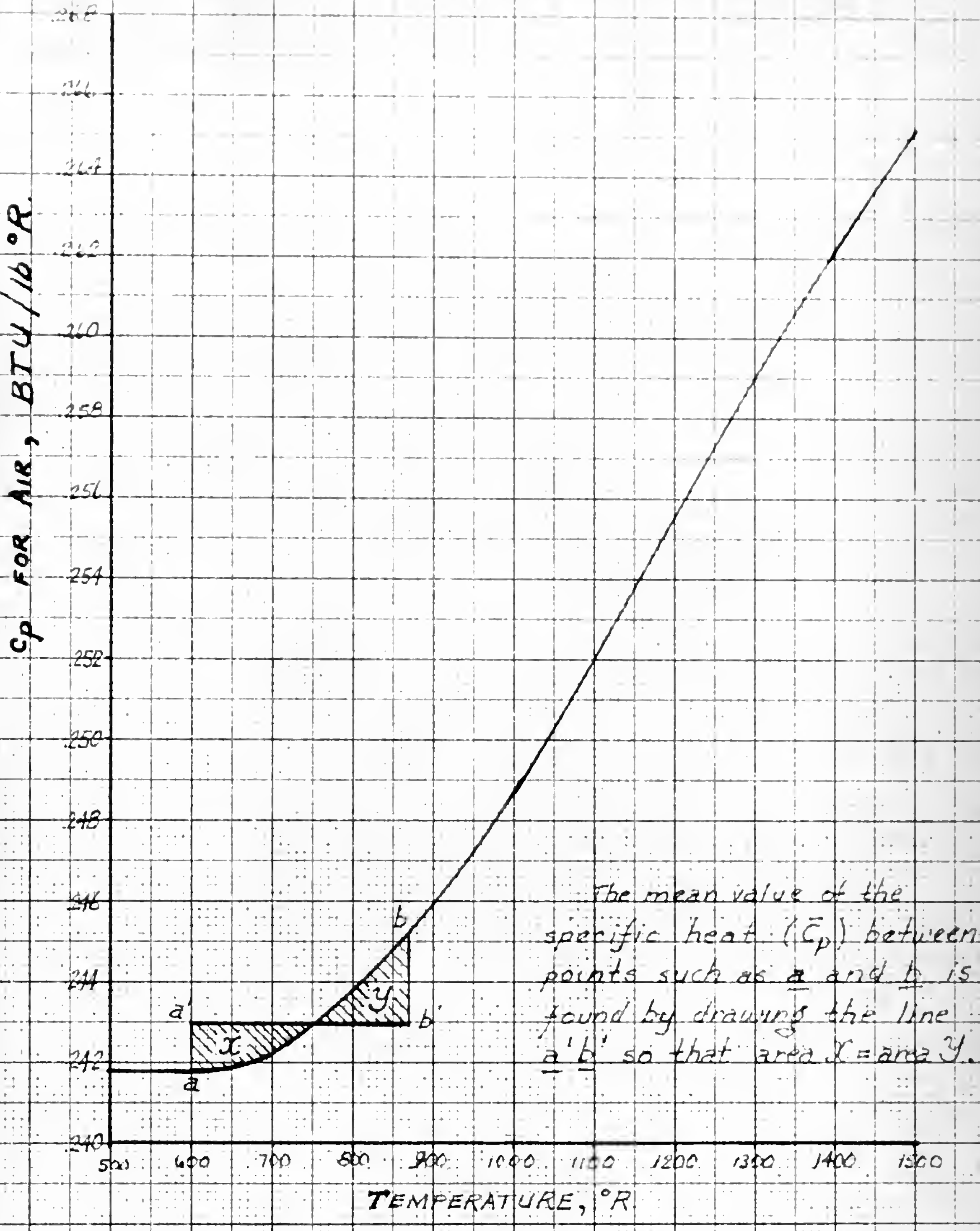


FIGURE 32. - DETERMINATION OF $\bar{C}_p$